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POLLUTANT CONTROL TECHNIQUES FOR PACKAGE BOILERS: HARDWARE MODIFICATIONS AND ALTERNATE FUELS

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16. ABSTRACT The report gives results of investigations of four ways to control nitrogen oxide (NOx) emissions from package boilers (both field operating boilers and boiler simulators): 1) variations in combustor operating procedure; 2) combustion modification (flue gas recirculation and staged combustion); 3) use of an alternate fuel, methanol; and 4) burner design optimization. Comparative tests were conducted in a firetube boiler. When firing natural gas, NOx emissions could be reduced to very low levels (about 20 ppm, corrected to 0% oxygen) by adding flue gas recirculation; the levels were less than those achieved with methanol without flue gas recirculation. As expected, flue gas recirculation was much less effective in reducing emissions from residual fuel oil, although emissions were reduced by 50% through the use of staged combustion. However, the same staging system increased NOx emissions from natural gas, indicating that further work is necessary to optimize the system for dual-fuel firing. Reduced excess-air firing was the only NOx control that both reduced emissions and increased thermal efficiency. Flue gas recirculation and staged combustion had negligible effects on boiler operating efficiency. Conversion to methanol results in a considerable decrease in thermal efficiency because of the water vapor's increasing the enthalpy of the flue gas.

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Boilers	Flue Gases	Stationary Sources	13A	
Burners	Fuel Oil	Package Boilers		
Nitrogen Oxides	Smoke		07B	
Combustion Control	Carbon Monoxide		21B	
Natural Gas			21D	

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ABSTRACT

Four methods of controlling NO_x emissions from package boilers have been investigated in both field operating boilers and boiler simulators. These control techniques were: 1) variations in combustor operating procedure; 2) combustion modification (flue gas recirculation and staged combustion; 3) the use of an alternate fuel, methanol, and 4) burner design optimization. Comparative tests were conducted in a firetube boiler. When firing natural gas, NO_x emissions could be reduced to very low levels (approximately 20 ppm corrected to zero percent oxygen) by the addition of flue gas recirculation which were less than those achieved with methanol without flue gas recirculation. As expected, flue gas recirculation was much less effective in reducing emissions from residual fuel oil although emissions were reduced by 50 percent through the use of staged combustion. However, the same staging system increased NO_x emissions from natural gas indicating that further work is necessary to optimize the system for dual-fuel firing. Reduced excess air firing was the only NO_x control technique which both reduced emissions and increased the thermal efficiency. Flue gas recirculation and staged combustion had negligible effects on boiler operating efficiency. Conversion to methanol results in a considerable decrease in thermal efficiency because of the water vapor increasing the enthalpy of the flue gas.

The staging system used in the firetube boiler tests was experimental and cannot be considered optimum. An alternative control option suitable for heavy fuel oil was tested in a watertube boiler simulator which would involve retrofitting with new burners. An oil-fired distributed mixing burner was tested with a high (0.77 percent) and a low (0.24 percent) nitrogen fuel oil. The largest reductions were obtained with the high nitrogen fuel (greater than 50 percent). Although this approach appears to be promising, further work is required to optimize atomizer designs and air velocities to improve preformance with respect to smoke and CO emissions.

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CONVERSION FACTORS

SI METRIC UNITS TO ENGLISH UNITS

To Convert From	To	Multiply By
kb/s	10^3 lb/hr	7.936640
ng/J	lb/10^6 Btu	2.326E-3
MJ/S	10^6 Btu/hr	3.412141
ug/J	lb/10^6 Btu	2.326
J/G	Btu/lb	4.299226E-1
MN/m^2	PSIA	1.450377E+2
KW/m^2	10^6 Btu/hr-ft^2	3.16998E-1

ENGLISH UNITS TO SI METRIC UNITS

To Convert From	To	Multiply By
10^3 lb/hr	kg/s	1.259979E-01
PSIA	MN/m^2	6.894757E-3
lb/10^6 Btu	ng/J	4.29922E+2
lb/10^6 Btu	ug/J	4.29922E-1
10^6 Btu/hr	MJ/S	2.930711E-1
Btu/lb	J/G	2.326
10^6 Btu/hr-ft^2	KW/m^2	3.154594

$^\circ F = 1.8\ (^\circ C) + 32^\circ$

1.0 SUMMARY

1.1 Program Objectives

Industrial boilers produce a significant fraction of the total nitrogen oxides (NO_x) emitted by stationary combustion sources. They are normally classified by their design type (watertube or firetube) and they range in size from 10×10^6 to 500×10^6 Btu/hr heat input. In the firetube design the combustion chamber and the convective tubes are surrounded either by water or steam and their maximum capacity (25×10^6 Btu/hr heat input) is dictated by the weight of water they must support. This limitation is avoided in the watertube design since the water flows through tubes which form the wall of the combustion chamber, thus reducing the total volume of water. The purpose of this study was to assess the various NO_x control options for packaged boilers. Packaged means that the boiler is completely preassembled by the manufacturer and shipped (normally by flatcar) ready for immediate installation. Packaged units are limited to 350×10^6 Btu/hr heat input and are oil or gas-fired.

Three different NO_x control options were considered in detail:

1. Flue gas recirculation - combustion products are withdrawn from the stack and added to the combustion air. This reduces the flame temperatures and the mean temperature within the boiler because the cooled combustion products act as thermal diluents.
2. Staged combustion - the combustion air supply is divided into two and the fuel reacts initially under oxygen-deficient conditions, then the second-stage air is added to complete the combustion process. Two different approaches to staging the combustion process are possible: (1) the fuel-rich zone is physically separated from the burnout zone and the two air supplies are added from different locations, and (2) the fuel-rich conditions are provided by reducing the rate of fuel/air mixing by burner design optimization. Both of these approaches were investigated.
3. Fuel switching - pollutant emissions can be minimized by converting to fuels whose characteristics do not promote NO_x formation. In this study methanol was evaluated as a potential fuel for packaged boilers.

In addition to these three options the influence of operational variables (load, excess air) on NO_x emissions were also investigated.

There have been a considerable number of investigations detailing the application of flue gas recirculation and staged combustion techniques to control the emission of nitrogen oxides from utility boilers. In comparison, very little is known concerning the practical aspects of applying these same techniques to package boilers. In a previous study(1), two package boilers operating in the field were modified to evaluate the effectiveness of both flue gas recirculation and staged combustion as pollutant control techniques for this class of equipment. The same boilers were used to provide further information on NO_x control options and on the impact of these control options on the thermal performance of the boiler. A preliminary evaluation of the EPA's burner concept for oil firing was carried out in a test furnace.

1.2 Facilities

Test Furnaces

Limited investigations were conducted in a laboratory firetube simulator in order to assess potential problems that might occur if methanol were to be used in industrial boilers. A special windbox design was fabricated and used to simulate a wide variety of burner aerodynamic conditions that might be encountered in practice. The combustor was designed to allow the use of both flue gas recirculation and staged combustion. The majority of the investigation was restricted to measurement of combustion product composition in the flue gases; however, some measurements were made of species distribution within the combustor.

The tests to establish the performance of oil-fired distributed mixing burners were carried out in test furnaces which were designed to simulate the conditions in water wall boilers. The small watertube simulator (SWS) was constructed to duplicate the combustion chamber of a typical D type boiler. The floor and firing wall were refractory; the remaining walls and roof were spray-cooled by water. The exit for the high-temperature combustion products was situated on a side wall duplicating the entry to the convective section of a conventional boiler.

Field Boilers

The major criteria for the selection of the boilers that were used in field investigations was that the tests should be carried out with two boilers of different design burning the same fuel oil. The two boilers tested were:

- a watertube boiler, 25,000 lbs/hr steam, and
- a firetube boiler, 12,000 lbs/hr steam.

Both boilers were modified to accept flue gas recirculation to the windbox. These modifications included a fan, ductwork and an automatic control system.

Staged combustion experiments were conducted only in the firetube boiler on an experimental basis. The only practical entry for the second-stage combustion air supply was through the rear of the firetube unit. Penetration of the front wall was rejected because it would necessitate cutting through 30 inches of refractory in the form of three cast refractory rings. The general arrangement of the staging system installed in the firetube boiler is shown in Figure 1-1. Ambient air was supplied by a separate fan to a distribution ring at the rear of the boiler. This ring supplied eight 2-inch stainless steel staging pipes which entered the firetube through the rear door. These pipes were laid along the wall of the firetube and provision was made to allow the axial location of the staging injectors to be varied in later epxeriments. The staged air was injected radially through fishtail orifices. Burner stoichiometry was varied by throttling the combustion air supply and maintaining a constant overall excess air by increasing the air flow through the staging injectors.

Oil-Fired Distributing Mixing Burner

A register burner has been developed under EPA sponsorship which burns pulverized coal in such a manner that NO_x emissions are minimized because the coal reacts with a fraction of the combustion air close to the point of injection. The staged air is delivered from discrete outboard staged air ports at the periphery of the burner exit. In this study an oil nozzle was fitted on the axis of the burner. When firing oil, the coal delivery pipe supplies only air. The burner design incorporates the following features:

- Three independently-controlled air streams to supply combustion air to:

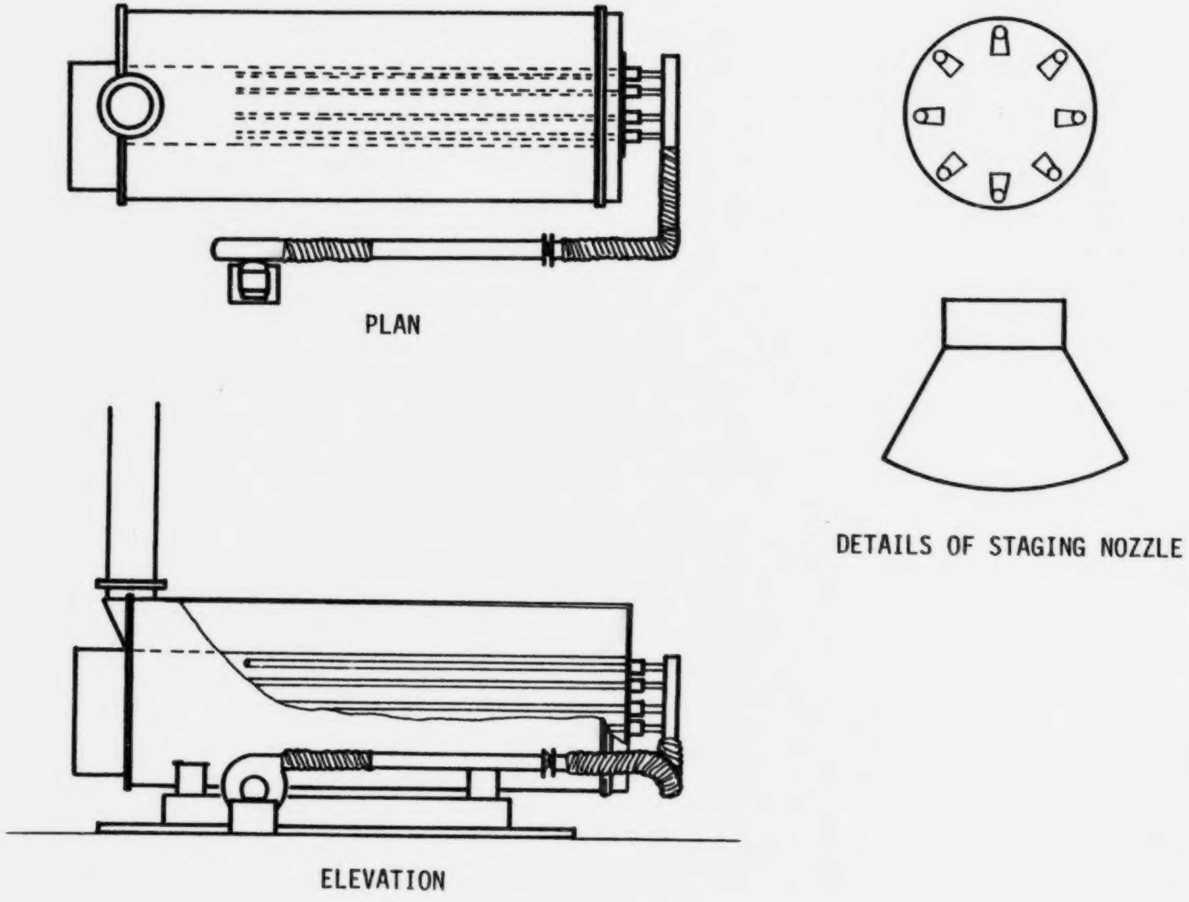

PLAN
ELEVATION
DETAILS OF STAGING NOZZLE

Figure 1-1. Details of Equipment Used in the Firetube Boiler Staging Investigation.

  - a primary channel;
  - a secondary channel;
  - outboard staged air ports.
- A retractable oil gun situated on the burner axis suitable for use with air or steam atomization.
- Interchangeable annular swirl vanes to provide swirl to the primary flow.
- Adjustable annular swirl vanes in the secondary stream.
- Interchangeable refractory exits. Two exits were used with length to throat diameter ratios of 1 and 2.

1.3 Results

Field Studies

Four control options have been investigated and compared on the basis of the tradeoff between changes in NO_x emissions and their effect upon thermal efficiency:

1. Reduced excess air-firing which may include the necessity for burner tune-up;
2. Fuel changes;
3. Flue gas circulation; and
4. Staged combustion.

Figure 1-2 presents a plot of NO_x emissions (ppm corrected to 0 percent O_2) against thermal efficiency which compares the influence of these various techniques for the firetube boiler. The CO limit shown represents the condition at which the CO concentration shows a marked increase from the baseline value. (Typically greater than 200 ppm). It can be seen that:

1. Reduced excess air firing will result in a decrease in both NO_x emissions and an increase in thermal efficiency. While no hardware modifications are required, this technique can only be considered as a control method for those boilers which are not operated at conditions giving maximum thermal efficiency unless burner modifications can result in even greater reductions in excess air levels.

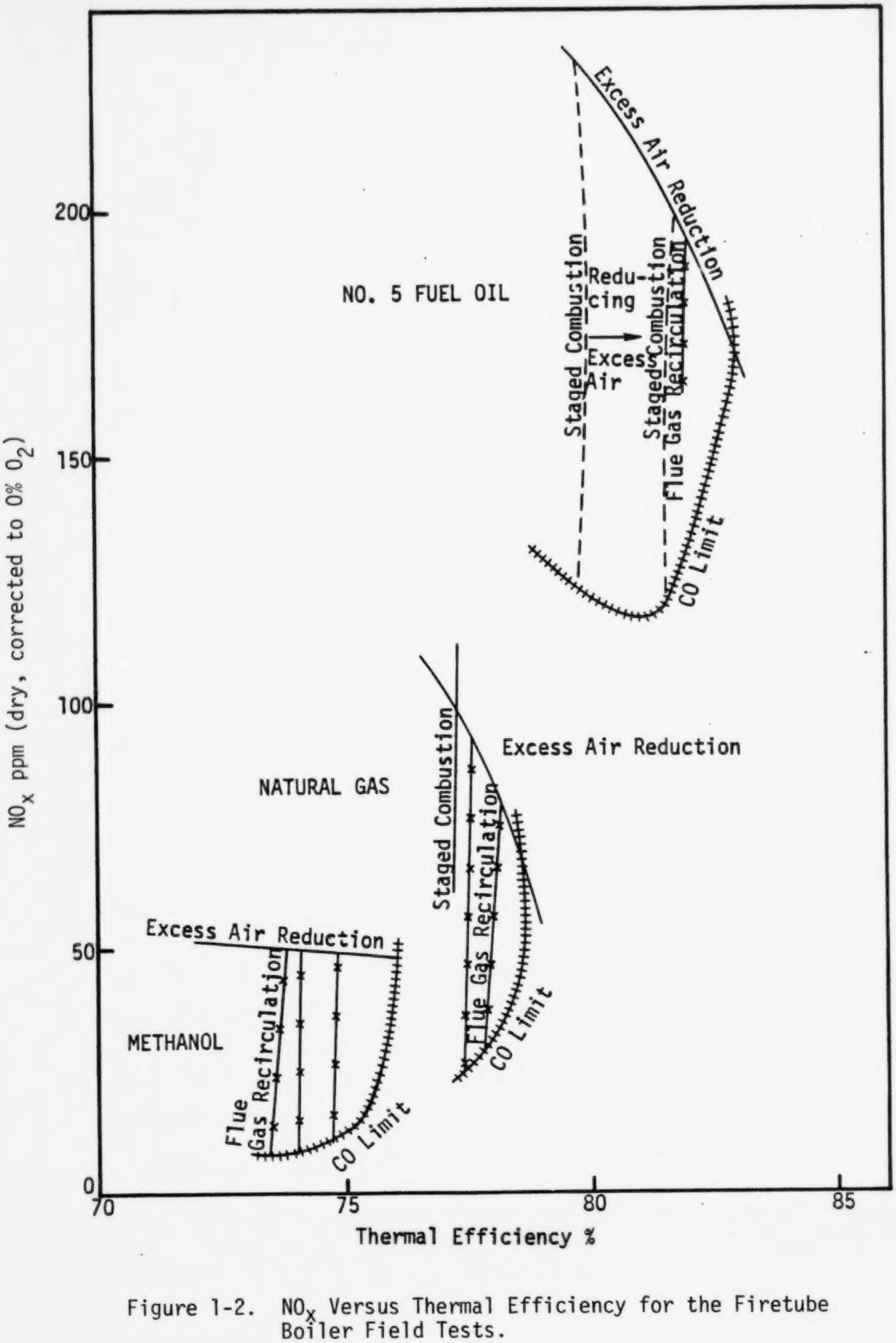


Figure 1-2. NO_x Versus Thermal Efficiency for the Firetube Boiler Field Tests.

2. Neglecting economic considerations, NO_x emissions would be reduced if methanol were to be used in preference to either natural gas or fuel oil in unmodified boilers. However, this decrease in NO_x emissions would also be accompanied by a decrease in thermal efficiency.

3. Flue gas recirculation does not have a strong influence on thermal efficiency (within the accuracy of the measurements, efficiency was unchanged with the addition of FGR), however, auxiliary power requirements increase. As a control technique it is most effective for gaseous and liquid fuels which do not contain bound nitrogen.

4. No influence of staged combustion on thermal efficiency was observed during the staging experiments. However, the staging equipment used in the firetube boiler was not nearly as effective with natural gas as with fuel oil.

The numerical values presented in Figure 1-2 relate to one boiler; however, the general characterisitcs and the observed trends will be similar for all boilers of this size. Due to the severe penalty in thermal efficiency, the use of methanol as a control technique to attain very low NO_x emissions can be discounted unless methanol costs are comparable to those of conventional fuels. Thus, significant reductions in NO_x emissions with increased thermal efficiency can only be achieved by combined application of staged combustion and/or flue gas recirculation with low excess air firing. The results presented in Figure 1-2 indicate two problems which require further consideration.

1. The use of staged combustion or flue gas recirculation optimized for maximum NO_x reduction, with one particular fuel in one boiler may not give optimum results with another fuel.

2. Staged combustion appears to be the better long-term approach, although reductions in NO_x emissions were limited to 50 percent with fuel oil firing and increased NO_x emissions were observed at some operating conditions with natural gas.

Evaluation of an Oil-Fired Distributed Mixing Burner

The staging system used in the firetube boiler tests was experimental and it cannot be considered optimum for watertube boilers. A much better control option would involve the use of a low NO_x burner which would require minimal alteration to the firing wall and windbox. The distributed mixing burner (DMB) could satisfy these needs. Based upon a preliminary evaluation, it appears that the oil-fired DMB would be suitable for use with high-nitrogen liquid fuels. Figure 1-3 compares the performance of the burner tested in this study with the correlation obtained by Cato, et al(3) based upon field tests of industrial boilers. It can be seen that the most spectacular reductions in NO_x emissions were obtained with the high-nitrogen fuel. Further work is needed to optimize atomizer designs and air velocities to improve the performance with respect to smoke and CO emissions.

1.4 Conclusion

Nitrogen oxides emitted by industrial boilers are produced from molecular nitrogen (thermal NO) and fuel bound nitrogen (fuel NO). For fuels which contain little or no fuel bound nitrogen flue gas recirculation is an effective method of reducing nitrogen oxide emissions. Residual fuel oils contain fuel nitrogen and the production of fuel NO is not strongly dependent upon temperature; therefore, flue gas recirculation does not provide a viable NO_x control option for this class of fuels. However, staged combustion techniques do minimize fuel NO formation but their effectiveness is most often limited by an increase in smoke emissions. The EPA's distributed mixing burner appears to have promise for application to high nitrogen liquid fuels.

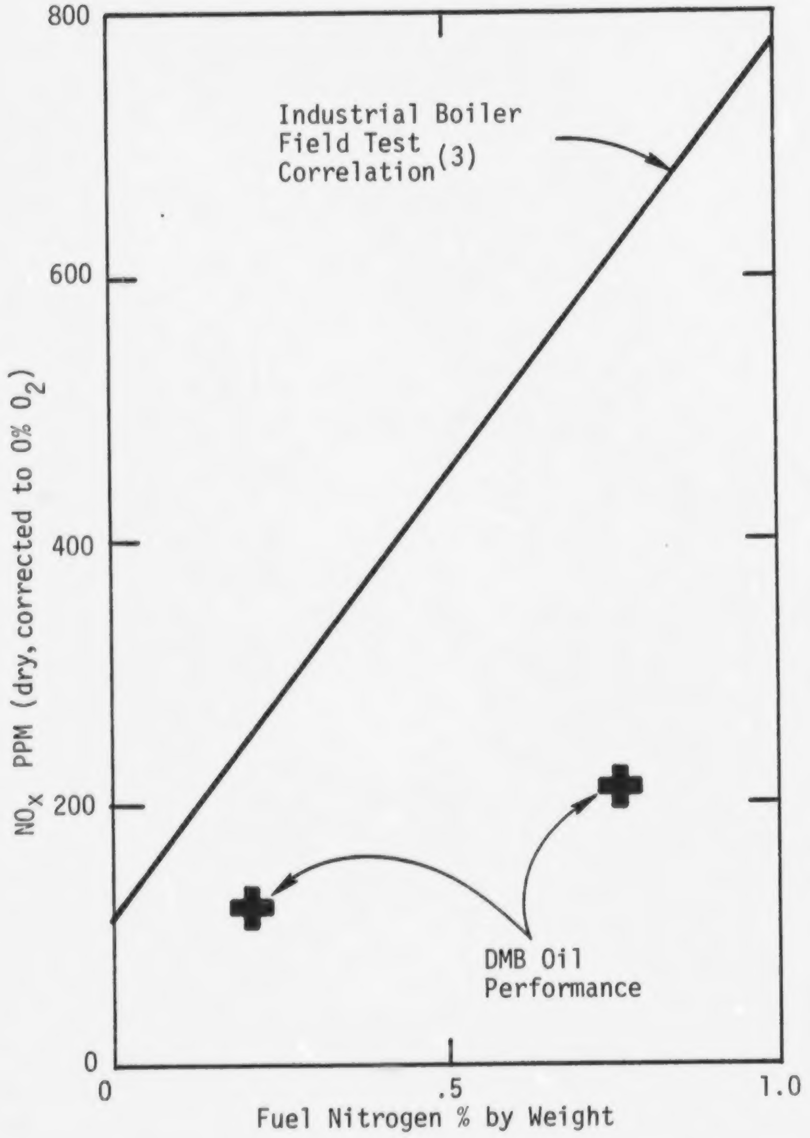


Figure 1-3. Oil-Fired DMB Performance.

2.0 INTRODUCTION

In a previous study, jointly sponsored by the EPA and the API (EPA Contract 68-02-0222) the effectiveness of various pollutant control techniques were evaluated for application to package boilers(1). The project involved laboratory investigations to determine the optimum use of flue gas recirculation and staged combustion for NO_x control from residual fuel oils. These laboratory investigations established the effectiveness of both techniques for residual fuels. Flue gas recirculation has only a minor influence upon NO_x formation from fuels containing bound nitrogen. Staged combustion has been demonstrated as a viable control technique but the level of the NO_x reduction is limited by the emission of excessive amounts of carbonaceous particulates. Modifications were made to two package boilers in the field to demonstrate the use of flue gas recirculation. The investigations were carried out on a watertube and a firetube boiler fitted at the same boiler site. In addition to the flue gas recirculation hardware, the firetube boiler was also modified to enable exploratory studies with staged combustion to be undertaken in an operational boiler.

The investigations described in this report were concerned with the further evaluation and optimization of various pollution control techniques and their influence upon the thermal efficiency of package boilers. Thus, this effort represents a logical continuation of the earlier studies that were almost exclusively concerned with fuel oil firing wherein the measurements were restricted to determining the influence of boiler modifications and operational variables on flue gas pollutant levels. In the present investigation, pollutant control techniques were evaluated for application with three fuels: natural gas, methanol and No. 5 fuel oil. Measurements were also made to enable the influence of these same control techniques on thermal efficiency to be assessed, thus ensuring that the control of one pollutant did not result either in the formation of other pollutants or a decrease in the effiency of fossil fuel utilization.

The other aspect of the work described in this report is the evaluation of methanol as a fuel for industrial boilers. This evaluation was restricted to the potential for pollution control and an assessment of possible practical problems associated with the utilization of alcohol fuels. It was not part of this study to access the economic impact associated with the production and distribution of these fuels. Considerable information is available describing the use of alcohol in internal combustion engines but there is a dearth of information in the public domain which compares the performance of boilers operated with conventional and alcohol fuels. In the present investigation, laboratory investigations and field tests were carried out to provide this comparison.

3.0 INDUSTRIAL BOILERS AS A SOURCE OF NITRIC OXIDE

Boiler firing accounts for greater than 50 percent of industrial fuel usage and almost all of this energy is used to generate steam or to heat water for process or space heating applications. Before discussing the pollutant emission characteristics of the two major industrial boiler types, it is worthwhile to discuss the existing industrial boiler population and possible future trends.

3.1 Source Characteristics

Industrial boilers are normally classified by their design type (watertube or firetube) and their size, which is normally described in boiler horsepower, steam generated per hour or hourly heat output (one boiler horsepower is approximately equal to 33.5 lbs of steam per hour or 33,500 Btu/hr heat output). Industrial size boilers range from 10×10^6 Btu/hr to 500×10^6 Btu/hr heat output and in the upper range they closely resemble utility boilers. Figure 3-1 presents data collected by Locklin et al(2) and shows the population breakdown of industrial boilers installed in 1972 by boiler type and capacity.

3.1.1 Firetube Boilers

Firetube boilers are used where the load is small and are limited in size to 25×10^6 Btu/hr heat input. As the name implies, the major characteristics of this design type is that the products of combustion pass through tubes surrounded either by water or steam. The major types of firetube boilers installed or currently available are:

- Scotch. Sales figures(2) show an increasing trend toward this type of boiler which was developed 30 years ago based upon the design of Scotch marine type boilers. One attractive feature of this design is its compactness, making it most suitable for locations with limited headroom. Maximum operating pressures are normally 350 psig, although some units are designed for 600 psig. Combustion takes place in a chamber with a diameter of 1 to 2 feet with L/D of up to 10:1, and the flame is normally tailored to fit this volume. The products of combustion exit the chamber and are directed through up to three tube passes. They can be either oil or gas-fired and are normally operated on a high/low firing cycle. Dry back boilers have a refractory lined rear wall, although more modern designs tend to be

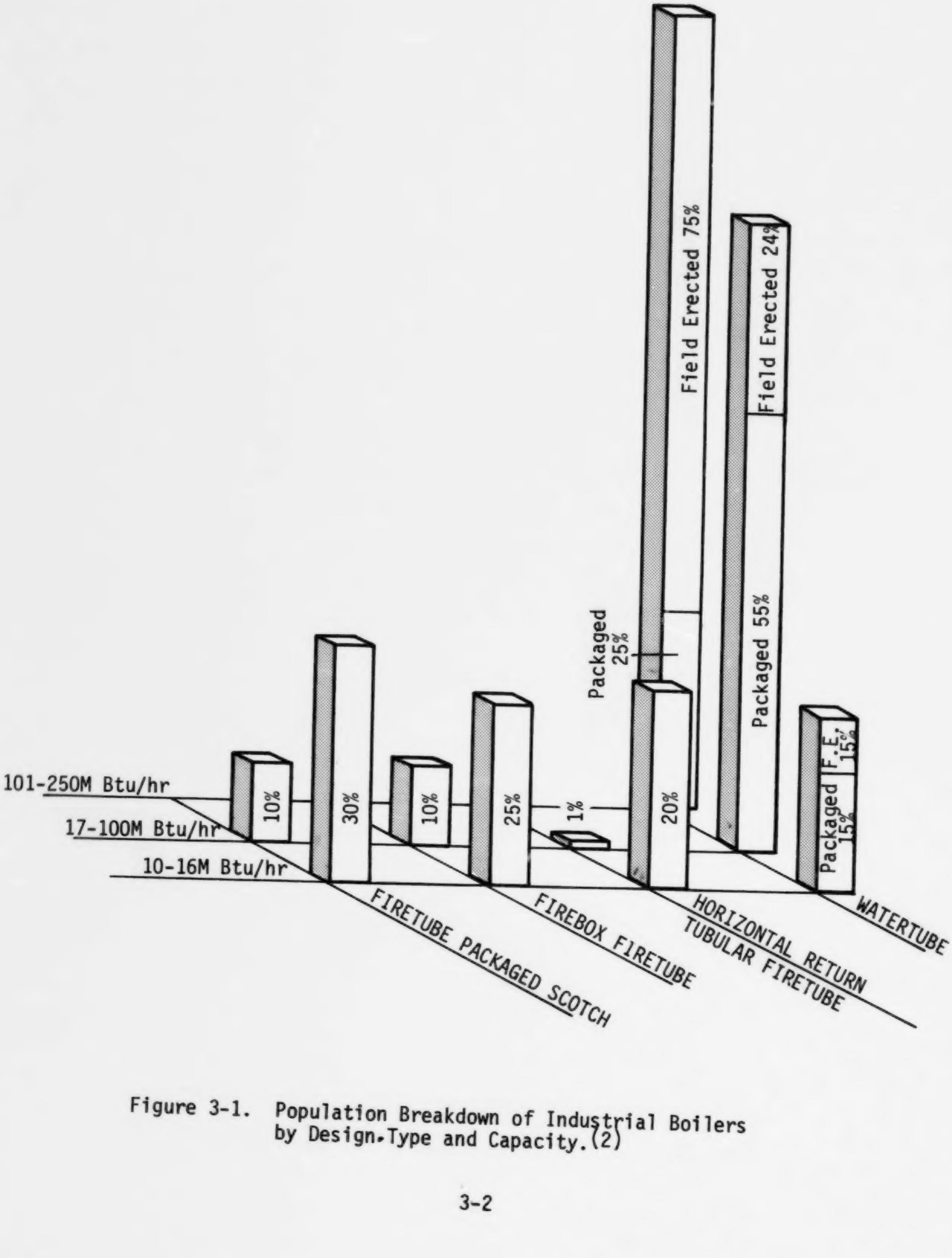


Figure 3-1. Population Breakdown of Industrial Boilers by Design Type and Capacity.(2)

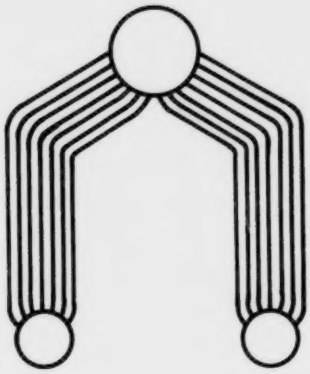
wet back (i.e., have a water-cooled rear wall). Every effort is made to maintain high heat transfer rates to reduce the heat transfer surface area. The gas velocities are maintained throughout the fire-tube passes by decreasing the diameter of the tubes in each pass. Turbulators are often fitted in the convective tubes to increase gas side heat transfer and internal baffling ensures good water circulation.

- Firebox Firetube. This type of design consists essentially of a horizontal shell with internal tubes and with a rectangular water-cooled furnace formed by water legs extending downward from the shell. They are normally limited to steam pressure of 200 psig and a capacity of 20×10^6 Btu/hr heat output.
- Horizontal Return Tubular. This was formerly the most popular type of firetube consisting of two passes, one under the shell and one through tubes. These boilers tend to have low thermal efficiencies.

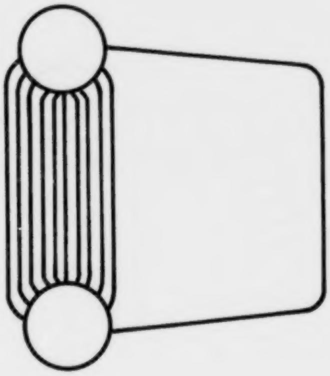
3.1.2 Watertube Boilers

The maximum capacity of firetube boilers is dictated by the weight of water they must support. This problem is avoided in watertube boilers since the water flows through tubes, thus reducing the volume of water. Also, this type of construction is more able to withstand high pressures than the firetube design and watertube boilers are normally used for steam pressures in excess of 150 psig. As indicated in Figure 3-1, watertube boilers can either be packaged or field erected. Packaged means that the boiler is completely preassembled by the manufacturer and shipped (normally by flatcar) ready for immediate installation. Packaged units are limited to 350×10^6 Btu/hr heat input and are oil or gas-fired because of the compact nature of the combustion zone. Field erected units may be coal-fired (stoker or pulverized) or fired by industrial or municipal waste.

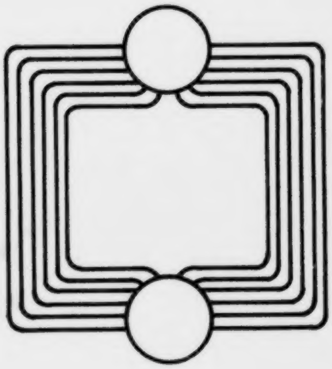
All watertube boilers manufactured today are of the bent-tube type. Straight watertube boilers have been superceded either by bent watertube boilers in the intermediate size, or by firetube boilers in the smaller size, although they may be important from retrofit aspects because many are still in service. Bent-tube boilers are classified by the number of drums, headers and tube configuration. The watertubes form the combustion chamber walls and are an integral part of the boiler; they are connected radially to the steam drum. The drum and heaters are arranged in three general configurations (see Figure 3-2). The larger



A-type has two small lower drums or headers. Upper drum is larger to permit separation of water and steam. Most steam production occurs in center furnace-wall tubes entering drum.



D-type allows much flexibility. Here the more active steaming risers enter drum near water line. Burners may be located in end walls or between tubes in buckle of the D, right angles to drum.



O-type is also a compact steamer. Transportation limits height of furnace, so, for equal capacity, longer boiler is often required. Floors of D and O types are generally tile-covered.

Figure 3-2. Drum and Header Configuration for Bent Tube Packaged Watertube Boiler.

industrial boilers (> 250 x 10^6 Btu/hr) resemble utility boilers and may be either tangential or wall-fired.

Forecasts indicate that by 1990, 99 percent to all watertubes up to 100 x 10^6 Btu/hr heat input will be packaged units as will 90 percent of the 101 to 250 x 10^6 Btu/hr(2).

3.1.3 Fuel Capability

Although this report is restricted to gas and oil-fired packaged industrial boilers, it is instructuve to compare the fuel capability of the industrial boiler population. As shown in Figure 3-3, the most popular fuels are natural gas and fuel oil, particularly in the small size ranges. Coal usage will undoubtedly increase in future years for field-erected watertube boilers. Another future trend that can be anticipated is the increased demand for dual-fuel capability. As natural gas supplies become restricted, more industrial consumers will be required to accept natural gas on an interruptible basis, thus necessitating dual-fuel capability.

3.1.4 Burner Characteristics

Burners may be classified both by their air delivery system and the method of fuel injection. Watertube boiler burners are generally of the register type. Until recently, the majority of burners used air registers to impart swirl to the air flow which promotes the formation of an internal recirculation bubble giving stable ignition. Figure 3-4a shows a dual-register burner with primary and secondary vanes for increasing the burner turndown capability. Figure 3-4b shows a venturi-type axial flow register. The recirculation zone needed for ignition stability is provided by the spinner vanes attached to the fuel injector. These vanes act as a self-cleaning bluff body, with most of the combustion air bypassing the vanes. Low particulate emissions at low excess air are claimed for this burner.

Gaseous fuels are usually supplied by multijet ring burners, whereas fuel oil may be atomized either mechanically or by steam or air. Figure 3-5 shows the distribution of oil atomizer types used in industrial boilers or different capacities. Air atomization is most prevalent in the smaller capacities but as boiler size increases, the use of steam predominates as it is more readily available. The use of rotary atomizers is restricted to the smaller size range, and it is estimated that by 1990 they will be completely replaced by steam or air atomizers.

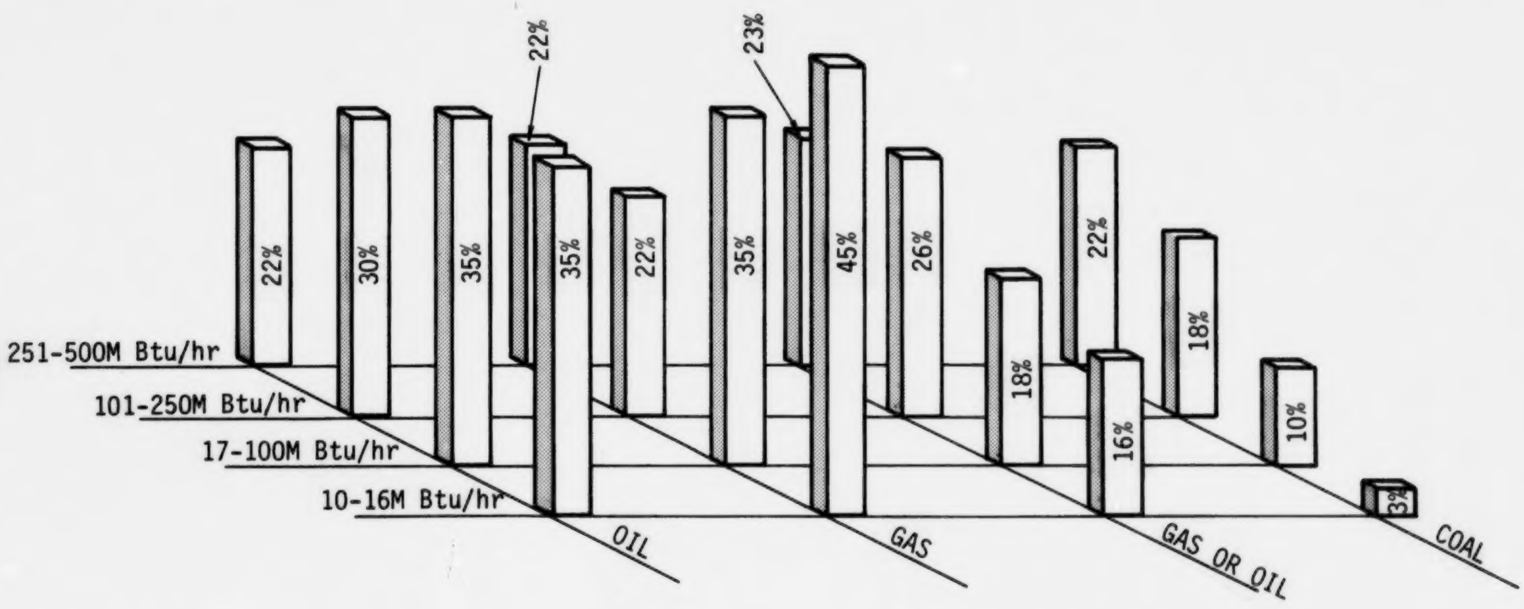


Figure 3-3. Population Breakdown of Industrial Boilers by Fuel Capability and Capacity.(2) (Rounded to nearest %, minor fuels not included.)

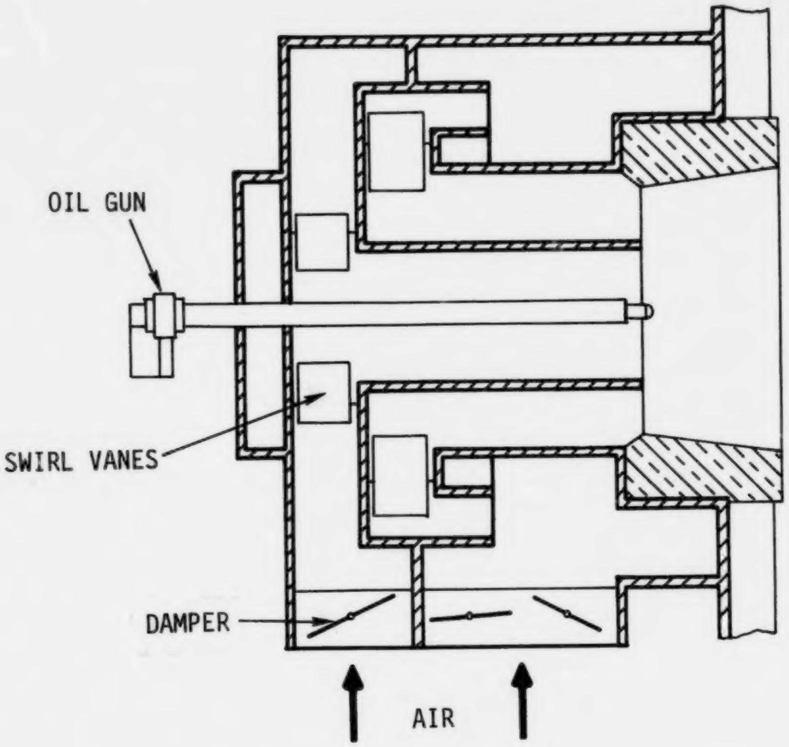

OIL GUN
SWIRL VANES
DAMPER
AIR

Figure 3-4a. Dual Register Oil Burner.

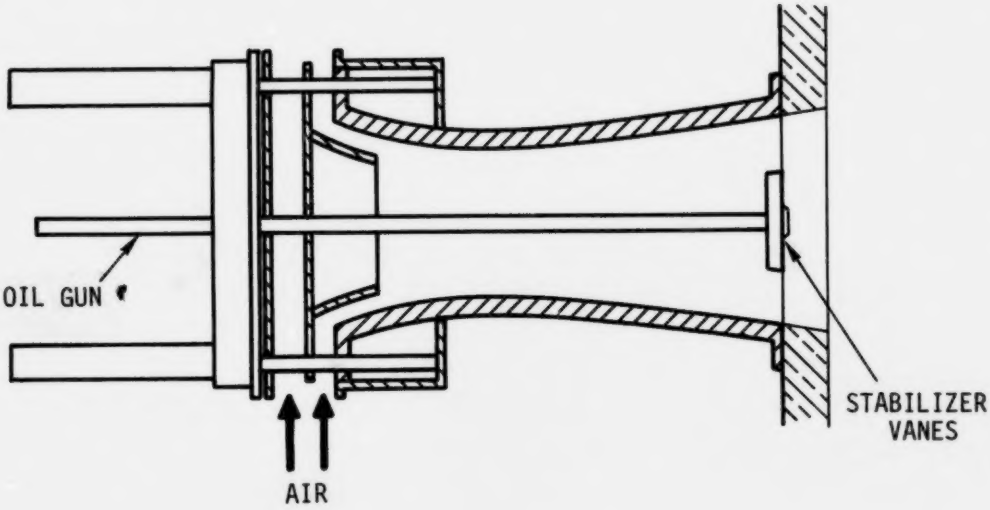

OIL GUN
AIR
STABILIZER VANES

Figure 3-4b. Venturi Type Axial Flow Register.

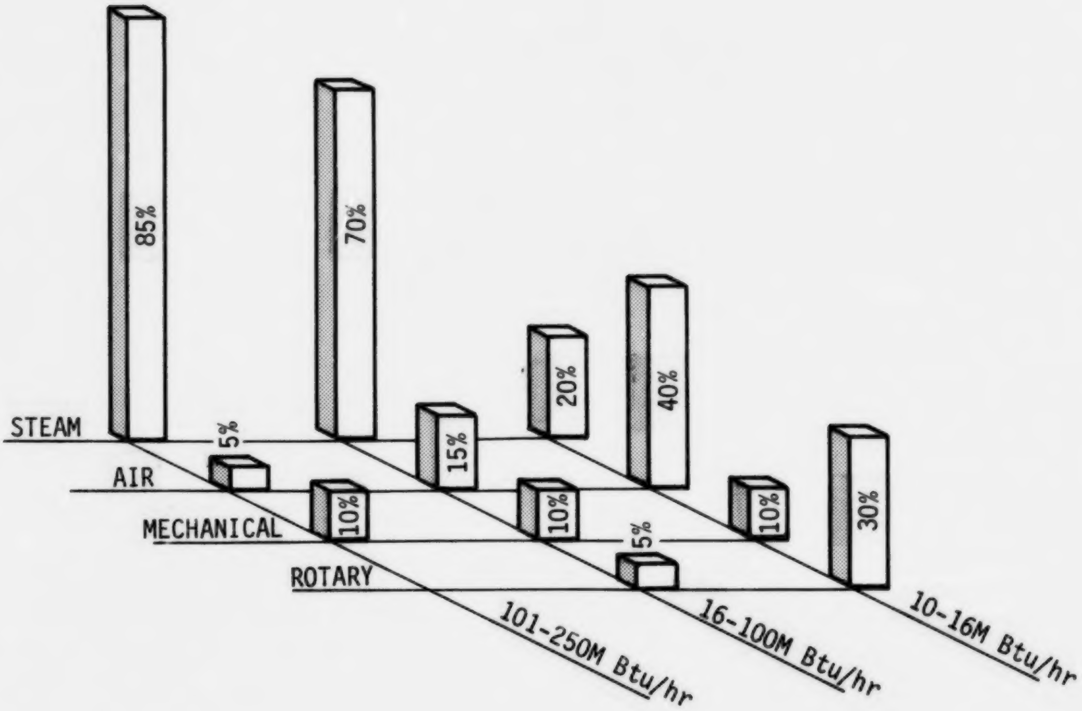


Figure 3-5. Population Breakdown of Industrial Boilers by Oil Atomization Method. (2)

3.2 NO_x Emission from Industrial Boilers

The most recent and authoritative study characterizing NO_x emissions from industrial boilers was carried by Cato et al(3) on behalf of the EPA. Approximately 50 boilers were tested and the characteristics of these boilers reflected the population distributions shown in Figures 3-1, -3 and -5. Cato et al also determined emissions for coal-fired boilers; however, the following discussions will be restricted to gas and oil firing. Figure 3-6 presents baseline data for the boilers tested during this study. Statistical analyses were carried out in an attempt to develop a general unifying theory describing the relationship between NO_x emissions and a set of boiler operational and design parameters for the various fuels. Multiple linear regression technique were employed and the following relationships evolved.

Natural Gas-Fired Boilers

$$NO_x = 12.4 \times \left(\frac{\text{80\% Boiler Capacity}}{\text{Number of Burners}}\right)^{.108} \times \left(\frac{\text{Furnace Volume}}{\text{Furnace Wall Area}}\right)^{.16} \times (\text{Air Preheat Temperature})^{.387}$$

Oil Fired Boilers

$$NO_x = \frac{701.5\ (\text{Number of Burners})^{.304} \times (\text{Fuel Nitrogen Concentration})^{.247} \times (\text{Furnace Volume})^{.109}}{(\text{Excess } O_2 \text{ Percentage})^{.751} \times (\text{Furnace Length})^{.455}}$$

These equations do not describe the behavior of a particular boiler, but rather describe the baseline emissions for the total set of industrial boilers. For natural gas firing, it is surprising that the excess oxygen concentration is not significant. Air preheat temperature appears to have a strong influence on baseline emissions for natural gas firing, but not for fuel oil firing. The dominant factors describing NO_x emissions from oil-fired boiler are fuel oil nitrogen content and excess air level.

3.2.1 The Influence of Boiler Design Parameters on NO_x Emissions from Industrial Boilers

Cato et al could find no systematic difference in the emissions from watertube or firetube boilers of similar capacities when burning the same type of fuel. Although furnace heat release rate did not shown a strong influence on emissions from either gaseous or liquid fuels, the most significant trend was observed with fuel oil; NO emissions increased as the heat release per unit

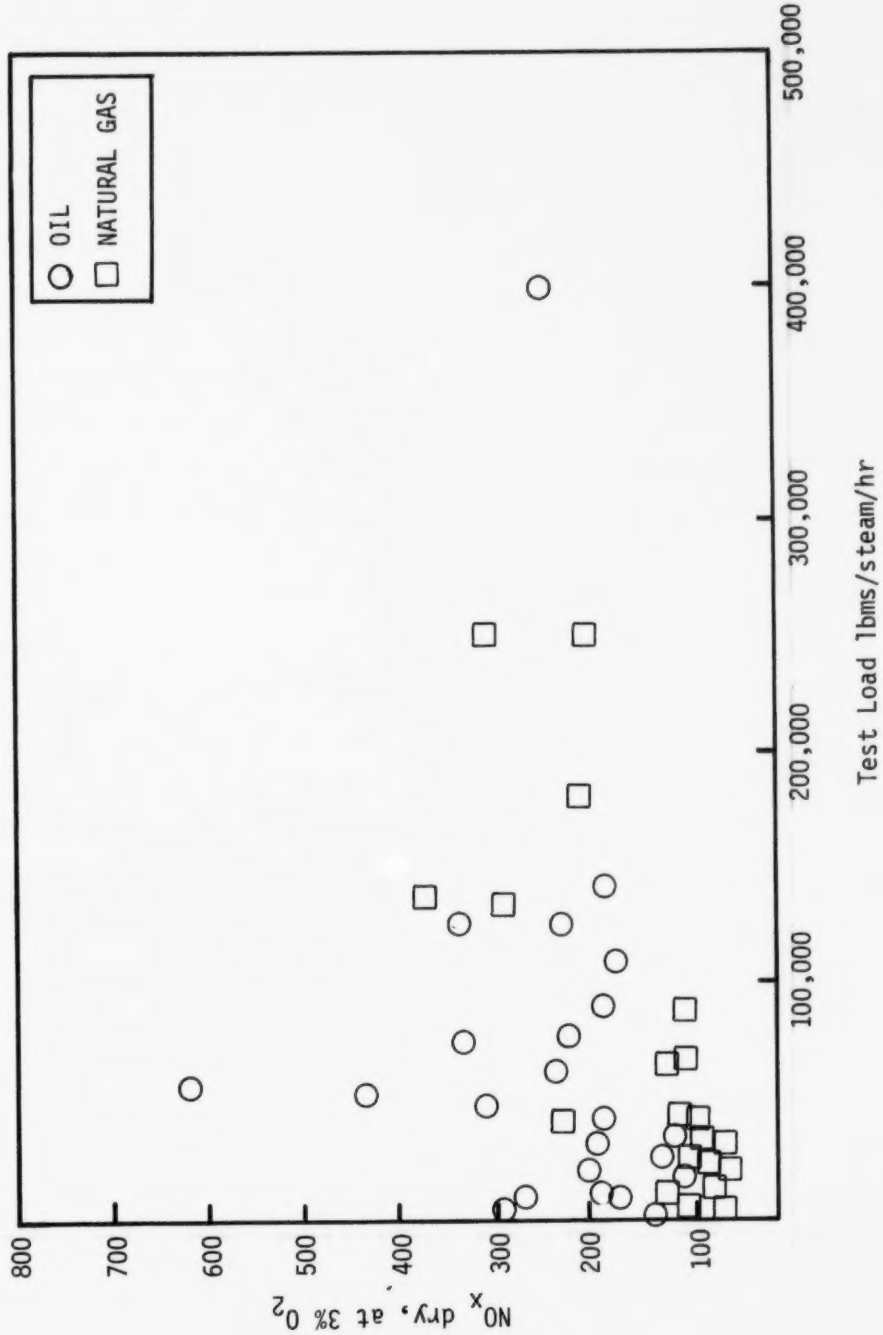


Figure 3-6. NO$_x$ Emission for Natural Gas and Oil-Fired Industrial Boilers as a Function of Boiler Size.(3)

volume increased. This observation is somewhat surprising since bulk gas temperature will tend to increase as combustion intensity increases which should cause an increase in thermal NO production. However, natural gas which produces only thermal NO shows minimal dependence of NO emissions on furnace heat release rate.

3.2.2 The Influence of Operational Parameters on NO_x Emissions

Low excess air firing is one of the methods used to reduce NO_x emissions from utility boilers. However, Cato indicates that for firetube boilers, NO_x emissions for both oil and gas firing are relatively insensitive to excess air. For watertube boilers firing oil, NO_x emissions were found to be strongly dependent upon excess air; however, with natural gas firing this dependency was found only when the air was preheated. As stated earlier, emissions from natural gas increased strongly as a function of air preheat, but data for oil firing indicates that air temperature does not strongly affect NO_x emissions with liquid fuels. NO_x emissions increased as the fuel oil temperature decreased. Boiler load did not have a strong influence on emissions except for natural gas firing with preheated combustion air.

3.2.3 The Influence of Fuel Properties on NO_x Emissions

The major fuel property affecting NO_x emissions is the quantity of bound nitrogen. Cato states that an average of 46 percent of the nitrogen in the fuel was converted to NO_x but does not indicate how the percentage conversion varied with the fuel nitrogen content.

3.2.4 The Influence of Burner Design on NO_x Emissions

Cato et al indicate that although the fuel oil conditions (e.g., temperature) influence NO emission, there is no effect of the atomizer type provided the fuel is efficiently atomized

3.3 The Potential for NO_x Control

Compared to utility boilers, industrial boilers, and particularly those below 100 x 10^6 Btu/hr heat output, have limited operational flexibility to reduce NO_x emissions. Most packaged boilers are single burner units, thus precluding the possibility of off-stoichiometric firing. Emission control options can be summarized as:

Operational Changes. Low excess air firing should reduce emissions from oil-fired units, but may have little influence on natural gas firing unless preheated air is used. As a short term measure, NO_x emissions could be reduced by converting from No. 6 fuel oil to natural gas which does not contain fuel bound nitrogen, and emissions are generally lower. In addition flue gas recirculation is a very effective control technique with natural gas.

Burner Redesign. Laboratory studies have indicated that NO_x can be reduced by burner redesign, but field tests have not indicated a preferential design.

Flue Gas Recirculation. Recirculation of cooled combustion products through the burner windbox is an effective method of reducing thermal NO_x formation, but it has little influence on fuel NO formation. Consequently, FGR is effective for natural gas-fired boilers, but much less effective for oil firing.

Staged Combustion. Staged combustion with interstage heat removal reduces both peak flame temperatures and oxygen availability and will thus allow the control of both thermal and fuel NO formation. The amount of control attained is dependent upon the amount of air staged, heat removed, location and method of secondary air introduction, distribution of fuel and air in the primary section, etc.

The use of these control options has been evaluated for (1) both methanol and conventional fuels under controlled laboratory conditions, (2) as operational conditions in the field, and (3) in terms of their control effectiveness and influence on thermal efficiency of boiler operation. Fuel results for both laboratory and fiel test data are presented in the subsequent sections.

4.0 METHANOL AS AN ALTERNATIVE FUEL FOR STATIONARY COMBUSTION SOURCES

One noticeable effect of Arab oil embargoes and the resulting fuel shortage has been an interest in the production of alternate fuels to replace traditional forms of energy such as natural gas and light oils. Methanol has been proposed as one alternative fuel. At present the major method of methanol production involves reforming methane to produce carbon monoxide, and then synthesizing a mixture of alcohols. U.S. natural gas is in short supply and therefore, this method of production does not represent a feasible source of methanol. Four other sources for methanol are being promoted by various groups(4):

- coal gasification to generate the synthesis gas;
- production from foreign natural gas an alternative to LNG transportation;
- production in Alaska as an alternative to natural gas pipelines from the North Slope; and
- biological synthesis.

Although the economics of methanol production as an industrial fuel must be considered in any comprehensive analysis such considerations are outside the scope of the present study. Rather the major objective of this report is the evaluation of the combustion characteristics of methanol in terms of pollutant emissions and thermal efficiency of steam generation.

A considerable amount of information is available concerning methanol combustion in reciprocating spark-ignition internal combustion engines,(5-12) however, only limited information is available pertaining to the combustion of these fuels in stationary sources. Duhl(13) has reported tests with methyl-fuels in both a small-scale combustor and in a utility boiler. Only limited results are reported, stating that in the small-scale test facility NO_x from methanol was 25-50 ppm compared to levels of 100-200 ppm for natural gas. Only qualitative results are available from the utility boiler tests stating the CO and UHC are insignificant, and that NO_x was less than that observed with natural gas. Martin(14,15) has carried out a series of investigations in a small laboratory test tunnel to compare NO_x emissions from alcohol fuels, distillate oil, distillate oil/water emulsion, alcohol/water mixtures and propane. Martin attempted to correlate the NO_x emissions as a function of fuel properties for various fuel and fuel mixes considered in his study. His results show:

- NO_x emissions correlated with the fraction of "equivalent" water content in different alcohols and distillate oil/water emulsions (the water content of an alcohol was considered as the prescribed part of the molecule). As the water content increases, NO_x emissions are reduced for comparable operating conditions.
- By vitiating the combustion air with cooled recirculated flue gases (FGR), NO_x emissions from distillate oil, propane, and isopropanol can be reduced below the NO_x levels obtained from methanol/air mixtures. For a given amount of FGR, the NO emissions from all four fuels are in the same relative order as the baseline emissions (distillate oil > propane > isopropanol > methanol).
- Although NO_x emissions could be correlated with the calculated well-mixed adiabatic equilibrium flame temperature for a given system, no overall correlation was found for (1) distillate oil with FGR, (2) distillate oil with water emulsion, (3) various alcohols and (4) alcohol/water emulsions.

Martin suggests that the low NO_x emissions from alcohol fuels are attributable to the "presence and level of oxygen in the fuel molecule, which can be viewed as a diluent carried in the fuel". The operative mechanism appears to be related to thermal considerations such as the latent heat of vaporization and/or the decreased flame temperature although no overall correlation with the well-mixed equilibrium flame temperature was obtained for all fuels. Martin does not rule out the possibility of chemical effects due to altered radical species concentration within the combustion zone.

Table 4-1 compares the physical properties of methanol (which accounts for greater than 90 precent of proposed commercial methyl fuels) and the appropriate properties for petroleum-based fuels. It can be seen that the heating value of methanol is much less than that of conventional liquid fuels. This will lead to immediate consequences for any system conversion in the reduction of both the effective capacity of the fuel storage system and the heat delivery rate of the fuel nozzle. The other physical properties of interest are the latent heat of vaporization and surface tension of the liquid, although it is difficult to do much more than specualte on their affect upon the combustion process. The vaporization of the liquid fuel droplets could have a significant influence on

Table 4-1. Fuel Properties

Property	Methanol	Heavy Liquid Petroleum (10-50° API)
Molecular weight	32	variable
Carbon to Hydrogen (weight ratio)	3.0	6 - 8
Carbon (weight percent)	37.5	84 - 87
Hydrogen (weight percent)	12.5	11 - 14
Oxygen (weight percent)	50.0	0.1
Specific Gravity (60°F/60°F)	0.796	1.0 - 0.78
Liquid density (lbm/gal at 60°F and 1 atm)	6.637	8.3 - 6.5
Coefficient of Expansion, 1/F at 60°F at 1 atm	0.00065	0.00035 - 0.00050
Viscosity, centipoises at 68°F and 1 atm	0.596	500 - 10,000
Specific heat of liquid, Btu/lbm-°F at 77°F and 1 atm	0.6	0.41 - 0.44
Heat of vaporization, Btu/lbm at 77°F and 1 atm	503	100 - 135
Heat of combustion, Btu/lbm at 77°F		
Liquid fuel - liquid H_2O	9776	18,100 - 19,800
Liquid fuel - gaseous H_2O	8593	17,400 - 18,800
Stoichiometric mixture, lbm air/lbm fuel	6.46	13.4 - 14.6

the local fuel/air temperatures in the vicinity of the vaporizing droplets. Methanol has a comparatively high latent heat of vaporization and taking account of the increased mass flowrate, approximately twelve times the amount of heat is required to vaporize methanol than distillate oil. Consequently, this will tend to reduce temperatures in the early stages of combustion. Martin(14,15) was able to correlate NO_x emissions from liquid fuels and fuel/water emulsions on the basis of a calculated latent heat of vaporization.

The other physical property which could have an influence on combustion characteristics and pollutant formation would be the liquid surface tension since this will influence liquid droplet size upon atomization. It can be considered that industrial turbulent diffusion flames contain "sheets" or clouds of droplets from which fuel vapor evolves for mixing and subsequent combustion, rather than a thin spray of droplets which burn individually. If this is true, then the major effect of liquid surface tension, and thus fuel droplet size, could be aerodynamic since the path of smaller droplets will be more strongly affected by the combustion of air flow field rather than the spray angle.

The formation of NO_x from the fixation of atmospheric nitrogen requires the rupture of nitrogen molecules which can occur by way of two general mechanisms: 1) attack of oxygen atoms, and 2) by hydrocarbon radicals such as CH and CH_2. The concentrations of oxygen atoms is strongly dependent upon temperature; thus the lower flame temperature of methanol in comparison to other fuels would account for the observed lower NO levels from methanol. An alternative explanation to the previous one of strictly thermal effects would be based upon the hypothesis that the oxidation of methanol results in a reduced oxygen atom overshoot or reduced CH or CH_2 concentration which would effect NO production. Calculations by Malte et al(16) for the NO formation in well-stirred systems shows a difference in NO between methanol and methane of a factor of about 2 and suggests that the differences cannot be attributed to oxygen atom kinetics.

5.0 EXPERIMENTAL FACILITIES

The evaluation of methanol as a fuel for industrial boilers was carried out in two different phases:

Phase I - Laboratory Investigations. A series of experiments were carried out to determine the combustion characteristics and the pollutant emission potential of methanol in a laboratory firetube boiler simulator.

Phase II - Field Tests. The pollutant emission characteristics and the thermal efficiency of two boilers were determined when they were burning No. 5 fuel oil, natural gas and methanol.

A description of the test equipment used during these investigations is presented below.

5.1 Firetube Simulator

The laboratory firetube was designed to simulate the conditions in the combustion chamber of a Scotch firetube package boiler. The design enable the use of both a commercial and an experimental burner, and the variation of input parameters such as fuel type, fuel load, atomizing medium, atomizing pressure, fuel/air mixing patterns, etc. The effectiveness of NO_x control techniques could be assessed as the system is equipped for recirculation of cooled flue gases through the burner (FGR) and staged combustion (SC) by downstream addition of secondary air.

5.1.1 Combustion Chamber

The 23.5 inch diameter combustion chamber consists of three 30-inch cylindrical sections and an end elbow section, all of which are cooled by Dowtherm fluid. Complete design details have been given elsewhere(17) and the salient features of the combustion chamber and the associated air and flue gas recirculation supply lines are shown in Figure 5-1. The heat absorbed by the cooling fluid can be measured in sections (two per cylindrical section and the elbow section) by monitoring the coolant temperature rise. Refractory inserts allow the effect of wall temperature to be investigated. However, this procedure also reduces the chamber diameter which will also influence the flow patterns and residence time in that section. The burners were joined to the combustion chamber by either of two burner exit sections, one of which was parallel and the other divergent (see Figure 5-3).

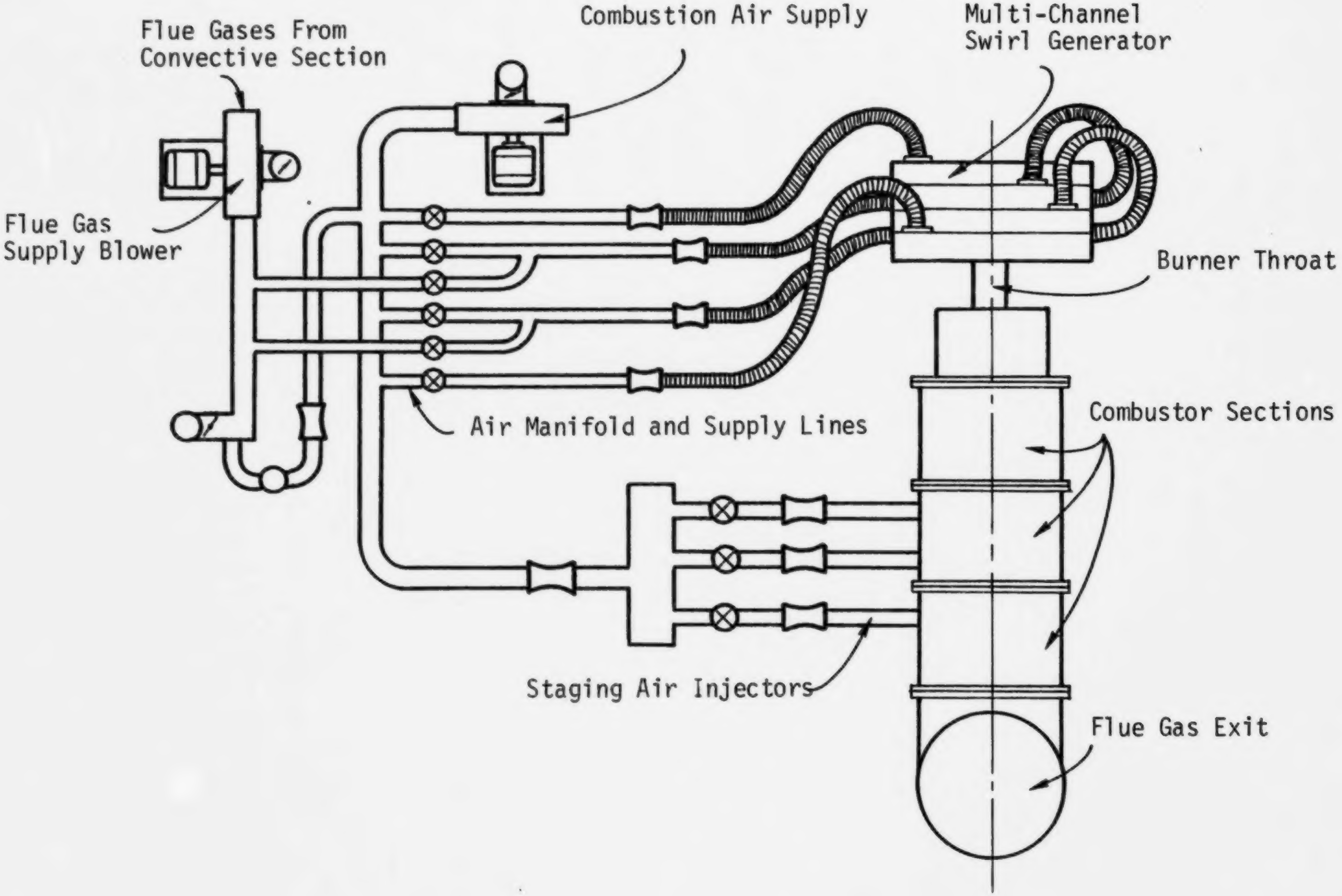

Flue Gases From
Convective Section
Combustion Air Supply
Multi-Channel
Swirl Generator
Flue Gas
Supply Blower
Burner Throat
Air Manifold and Supply Lines
Combustor Sections
Staging Air Injectors
Flue Gas Exit

Figure 5-1. Schematic of Laboratory Firetube Simulation.

5.1.2 Burner Designs

Two difference burners were used throughout the laboratory investigations. One was a modified commercial burner which had been used in earlier investigations(17). The second burner was purely experimental and was designed to allow a wide range of air/fuel mixing patterns to be investigated.

The Commercial Burner

In Figure 5-2, a sketch is presented showing the salient features of a modified 3.7 x 10^6 Btu/hr Ray Burner. The two major modifications which distinguish the burner used in the laboratory investigations from those used in the field are:

- the air supply to the primary and secondary sections were physically separated to provide for independent control and monitoring of each air stream;
- the fuel flowrate and total airflow were uncoupled to allow independent variation of each parameter.

The primary air flow is axial but some rotation is imposed upon the secondary air stream by the inlet air damper. The burner is designed to fire natural gas, fuel oil or combinations of both fuels. Oil is injected by a low pressure twin fluid atomizer which can be operated with either air or steam.

The Experimental Burner

The experimental burner was designed to fulfill two objectives:

- it must provide for a wide variation in the distribution of both axial and tangential velocity at the burner throat; and
- it must be adaptable to several different exit geometries.

The air distribution at the burner throat was controlled by a multi-channel swirl generator (MCSG) which is shown in Figures 5-3 and 5-4.

Air is supplied to four annular ducts through four separate chambers. Combustion air from the delivery manifold enters each chamber through a 90^0 elbow section. By rotation of the elbow section, as shown in Figure 5-4, the velocity vector of the combustion air may be directed at the axis of the burner or at a distance off-center from the axis. With the velocity vector directed at the axis, no tangential momentum is imparted to the air and the resulting air in the

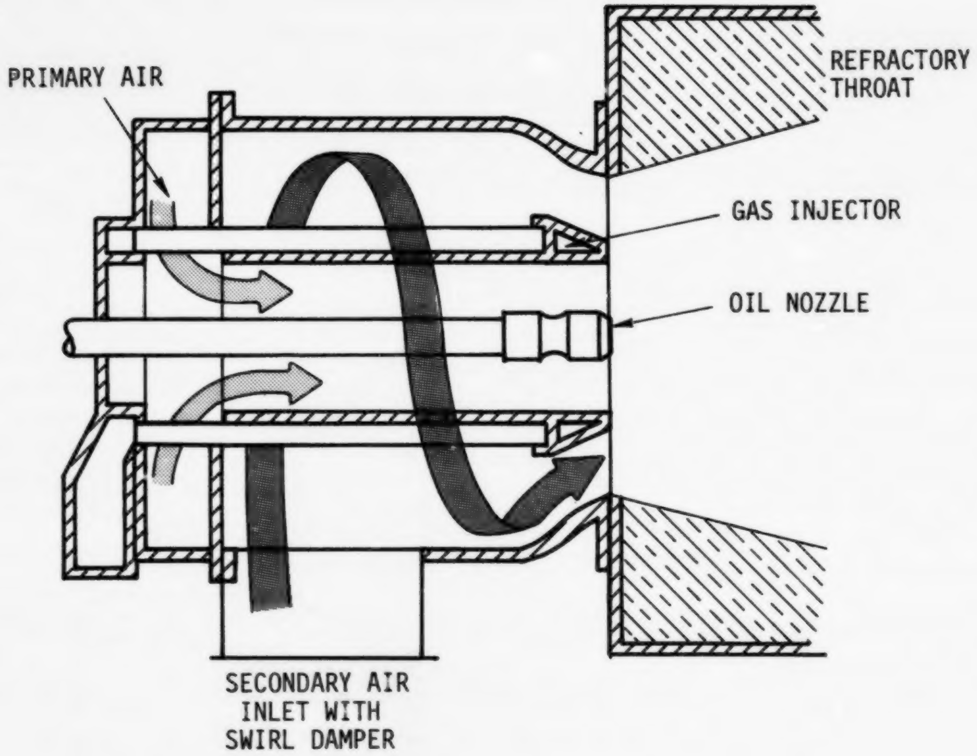
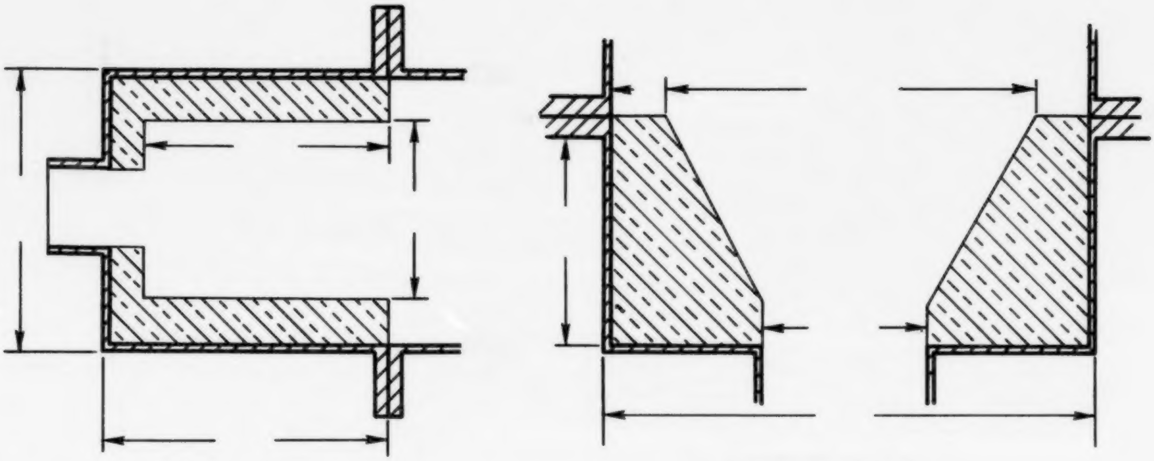

PRIMARY AIR
REFRACTORY THROAT
GAS INJECTOR
OIL NOZZLE
SECONDARY AIR INLET WITH SWIRL DAMPER

Figure 5-2. Sketch of Modified Commercial Burner Used in the Laboratory Investigations.



PARALLEL EXIT DIVERGENT EXIT

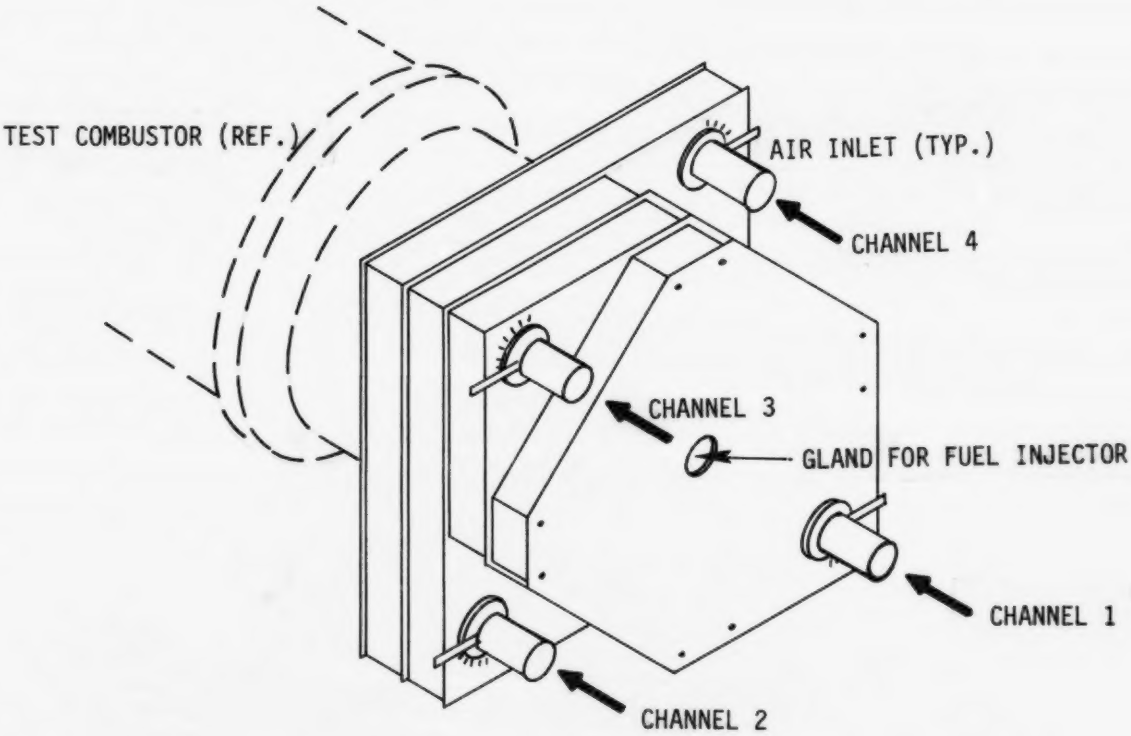

TEST COMBUSTOR (REF.)
AIR INLET (TYP.)
CHANNEL 4
CHANNEL 3
GLAND FOR FUEL INJECTOR
CHANNEL 1
CHANNEL 2

Figure 5-3. Multichannel Continuously Variable Swirl Generator - Rear View and Two Exit Geometries.

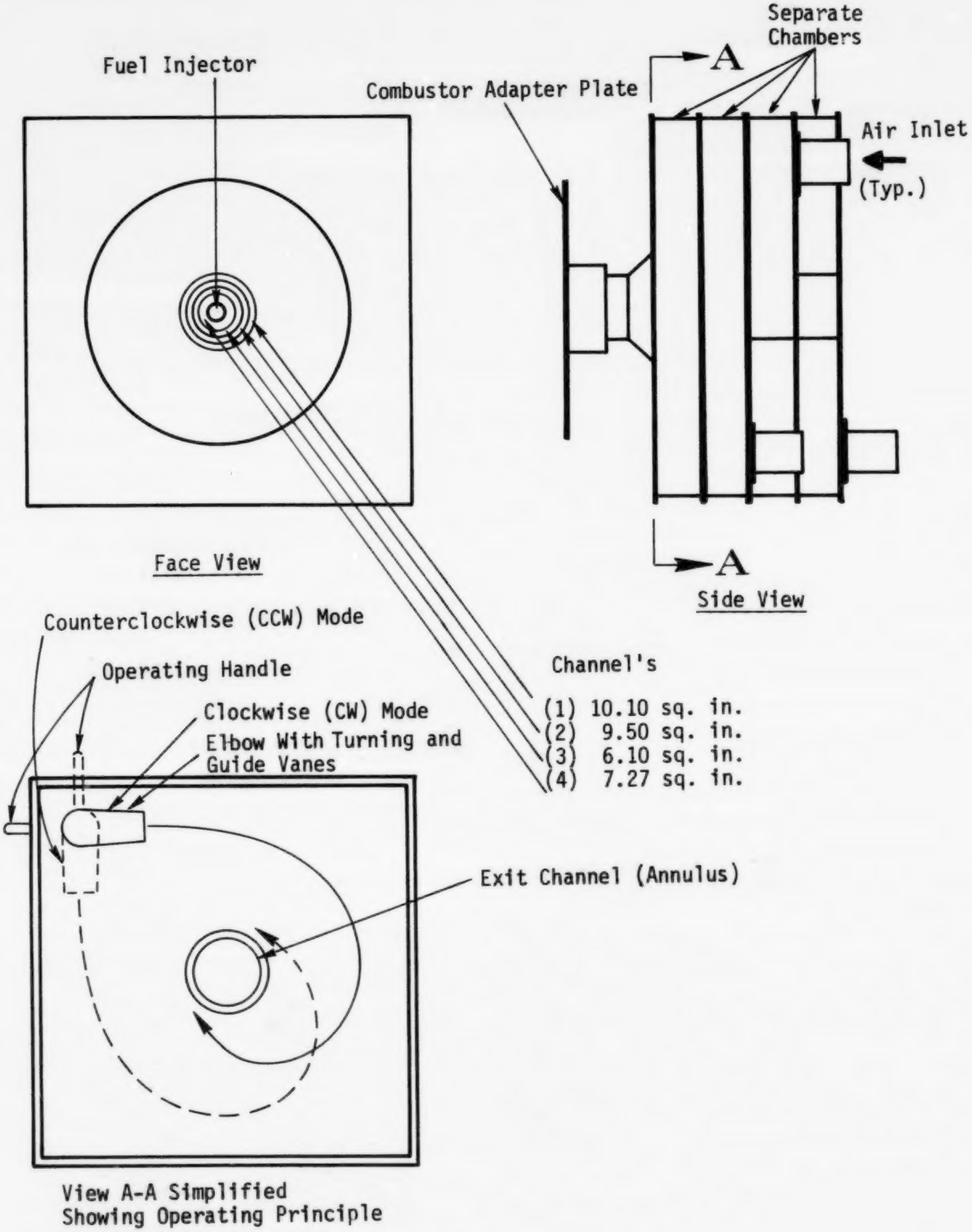

Separate Chambers
Fuel Injector
Combustor Adapter Plate
A
Air Inlet
(Typ.)
A
Face View
Side View
Counterclockwise (CCW) Mode
Operating Handle
Channel's
Clockwise (CW) Mode
Elbow With Turning and Guide Vanes
(1) 10.10 sq. in.
(2) 9.50 sq. in.
(3) 6.10 sq. in.
(4) 7.27 sq. in.
Exit Channel (Annulus)
View A-A Simplified
Showing Operating Principle

Figure 5-4. Multichannel Continuously Variable Swirl Generator - Details.

annular section is completely axial. With the velocity vector directed off-center, a rotational flow is developed within the chamber, thus the resulting annular flow has both axial and tangential velocity components. The toal axial/tangential momentum ratio of the annular flow is determined by the amount of nozzle rotation and the proportion of total combustion air in each chamber.

The air flow pattern created by the MCSG is specified by the percentage of air flow and inlet nozzle rotation in each chamber. A specification of a 20/20/30/30 air distribution indicates the percentage of the total combustion air in air channels 1/2/3/4, respectively. Likewise a swirl configuration of 20/20/40/40 indicates the degree of rotation of the inlet air nozzles 1/2/3/4, respectively. The air/swirl configuration 20/20/30/30, 20/20/40/40 uniquely defines the test conditions of the MCSG.

5.1.3 Combustion Modification Facilities

Flue gas recirculation (FGR) is made possible by withdrawing combustion production gases from the stack and cooling them before addition to the air delivery manifold, Figure 5-1. The cooled combustion products could either be mixed with the total combustion air or divided into one or two streams for injection into a specific location in the burner. Staged combustion (SC) is possible by diverting a portion of the combustion air into any one or combination of three sets of air injectors at three axial locations from the burner exit. Each axial station consists of four injectors and the injectors may be oriented such that the second stage air is directed upstream, downstream, or tangential to the bulk flow of combustion gases in the combustion chamber.

5.1.4 Measurement Techniques - Laboratory Investigations

The laboratory combustor was instrumented to provide necessary data for boiler performance and assessment of pollutant emissions. The monitored parameters for the input/output investigations were:

Input

Air flow
Fuel flow
Fuel temperature
Atomizing fluid pressure

Output

Individual section coolant temperature rise
Flue gas temperature
Flue gas species composition

The flue gas composition was measured with the following instruments:

Oxides of Nitrogen - Beckman Model 951 Chemiluminescent Analyzer

Carbon monoxide - Beckman Non-Dispersive Infrared (NDIR) Analyzer for low concentrations (10-10,000 ppm), Horiba Instruments Mexa-300 NDIR for large concentrations (0.5-15 percent).

Carbon dioxide - Horiba Instrument Mexa-300 NDIR.

Oxygen - Taylor Servomex Paramagnetic Analyzer (Type OA251)

Unburned hydrocarbons - Beckman Model 402 Flame Ionization Detection (FID) Hydrocarbon Analyzer

Partially oxidized hydrocarbons (aldehydes) - an independent testing laboratory was used to measure aldehydes by a wet chemical method

Smoke - Smoke readings were taken with a Bacharach Smoke Scale Tester.

Certified zero and calibration gases were used to calibrate these monitors throughout the investigation. Frequent calibration checks were used to ensure the reliability of the data.

5.2 Facilities for Field Investigations

5.2.1 The Watertube Boiler

The watertube boiler was built by Superior Combustion Industries and was designed to fire No. 5 fuel oil or natural gas. The boiler is fired by a register burner, a schematic of which is shown in Figure 5-5. The natural gas is injected from a gas ring and the oil gun utilizes steam atomization. The front wall of this "D" type forced draft unit is refractory lined and the gas passages are horizontal. The combustion gases exit the radiant section, turn through 180 degrees before passing through the convective section. The unit is rated at 25,000 lbs of steam per hour at 250 psig, and no provision is available for air preheat.

The flue gas recirculation system was designed to recirculate 30 percent

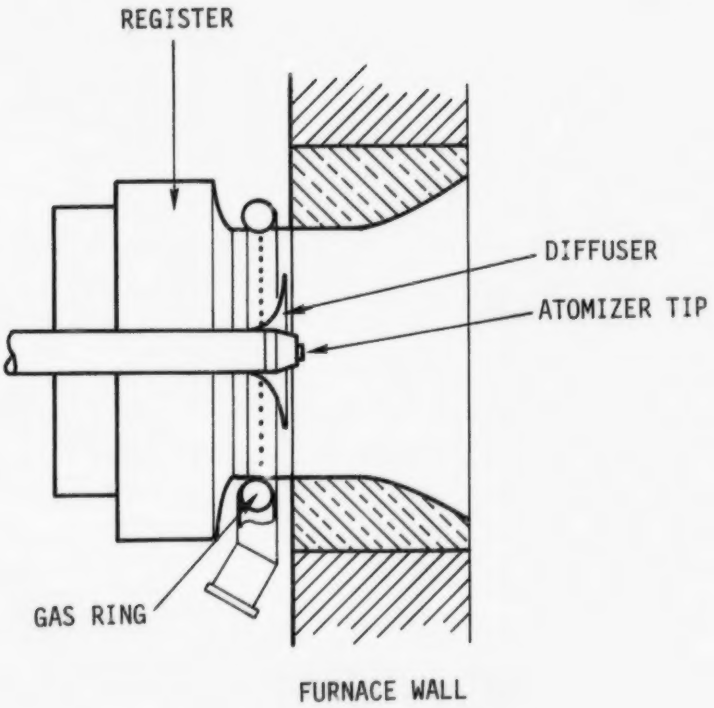

REGISTER
DIFFUSER
ATOMIZER TIP
GAS RING
FURNACE WALL

Figure 5-5. Register Burner Installed in the Watertube Boiler.

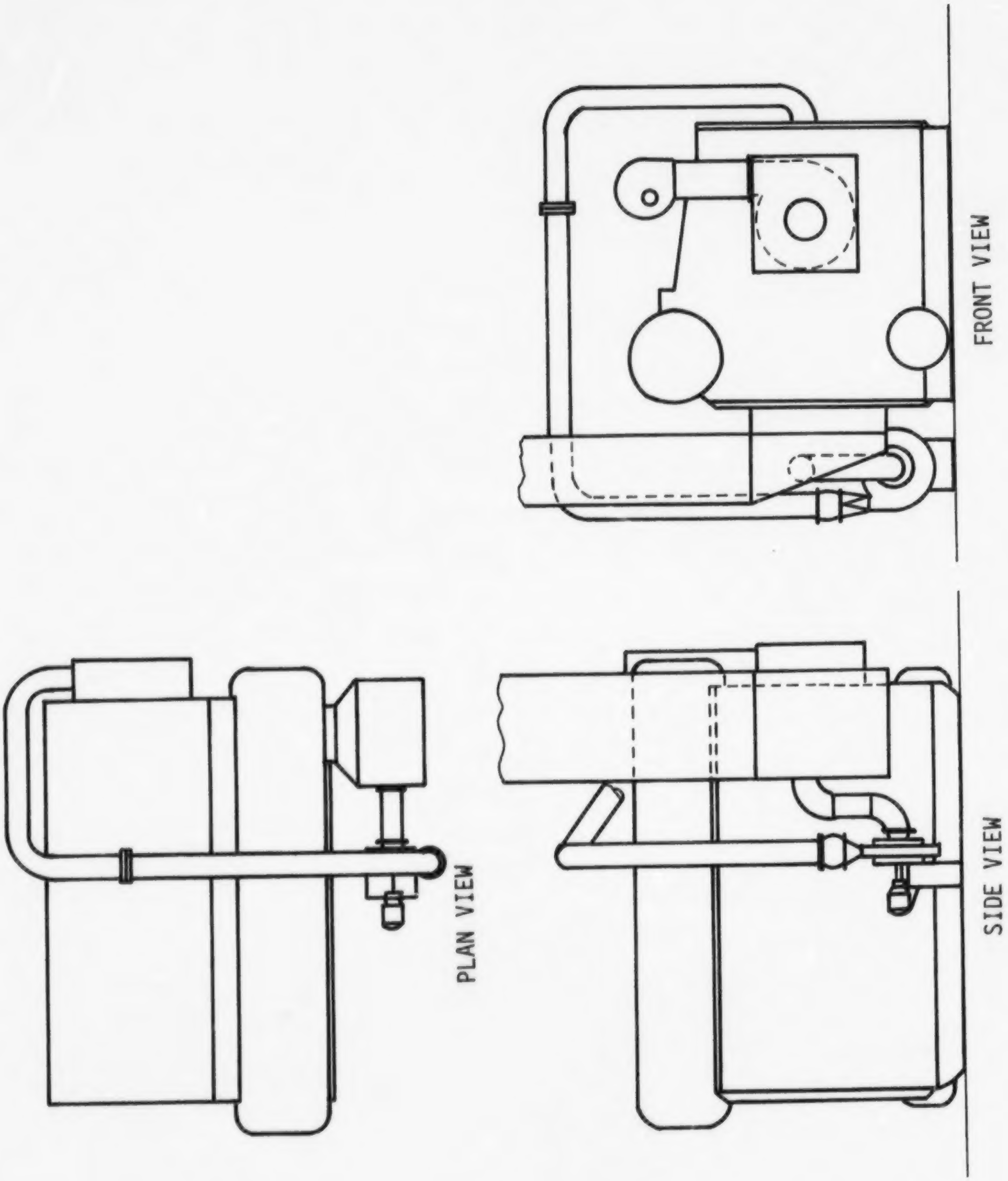

FRONT VIEW
SIDE VIEW
PLAN VIEW

Figure 5-6. Layout of Flue Gas Recirculation System for the Watertube Boiler.

of the full load combustion products through the windbox. The system installed in the watertube boiler is shown in Figure 5-6. The recirculation fan was located at grade level between the two watertube boilers adjacent to the stack breeching for both boilers. The flue gases were withdrawn before the boiler breenching dampers and the 12-inch ID ducts carried the gases over the boiler before entering the windbox normal to the downward flow of the combustion air. The ductwork was fabricated from rolled No. 10 gauge carbon steel plate with longitudinally welded seams. The mass of recirculating combustion products was controlled by a damper installed at the immediate discharge of the fan and metered by an ASME standard orifice installed in the horizontal duct passing over the boiler.

5.2.2 The Firetube Boiler

The application of both flue gas recirculation and staged combustion control techniques were investigated in the firetube boiler which was also manufactured by the Superior Combustion Industries and a sketch of the burner firetube arrangement is shown in Figure 5-7. The boiler was designed to burn No. 5 fuel oil with an air atomized tip. Natural gas can also be burned and it is injected from a ring embedded in the refractory throat and the boiler is refractory lined for the first 2.5 feet. The unit has four gas passes, one radiant and three convective passes. The firetube and second pass are surrounded by water and the third and fourth passes lie in the vapor space at the top of the boiler. The forced draft unit is rated at 12,000 lbs of steam per hour at 250 psig.

Details of the flue gas recirculation system installed in the firetube boiler are presented in Figure 5-8. All duct work was 10-inch ID and the control and metering system was similar to that used on the watertube boiler. The recirculation fan was placed on the grating above and to the rear of the boiler. Combustion gases were withdrawn from the stack at this level and entered the windbox tangentially in the same flow direction as the combustion air from the F.D. fan. When the windbox was breeched to accept the recirculation ductwork, a baffle plate, not detailed on any Superior drawings, was found in the windbox. This baffle plate reduced the volume of the windbox and also obstructed the reicrculation gas entry. Since at that time, no function could be attributed to this plate, it was removed. Subsequent experience appeared to indicate that this was an error since it was necessary to replace most of the baffle to

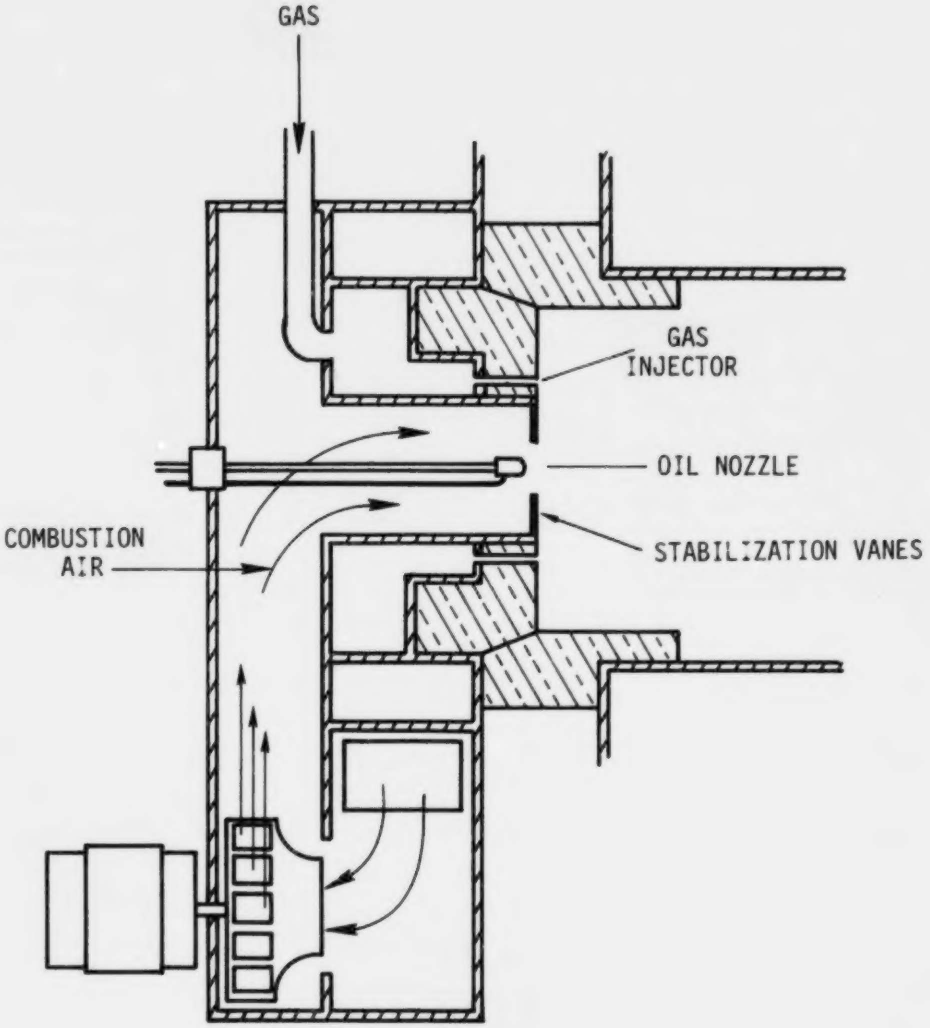

GAS
GAS INJECTOR
OIL NOZZLE
COMBUSTION AIR
STABILIZATION VANES

Figure 5-7. Sketch Showing the Windbox Burner Arrangement of the Firetube Burner (Not to Scale).

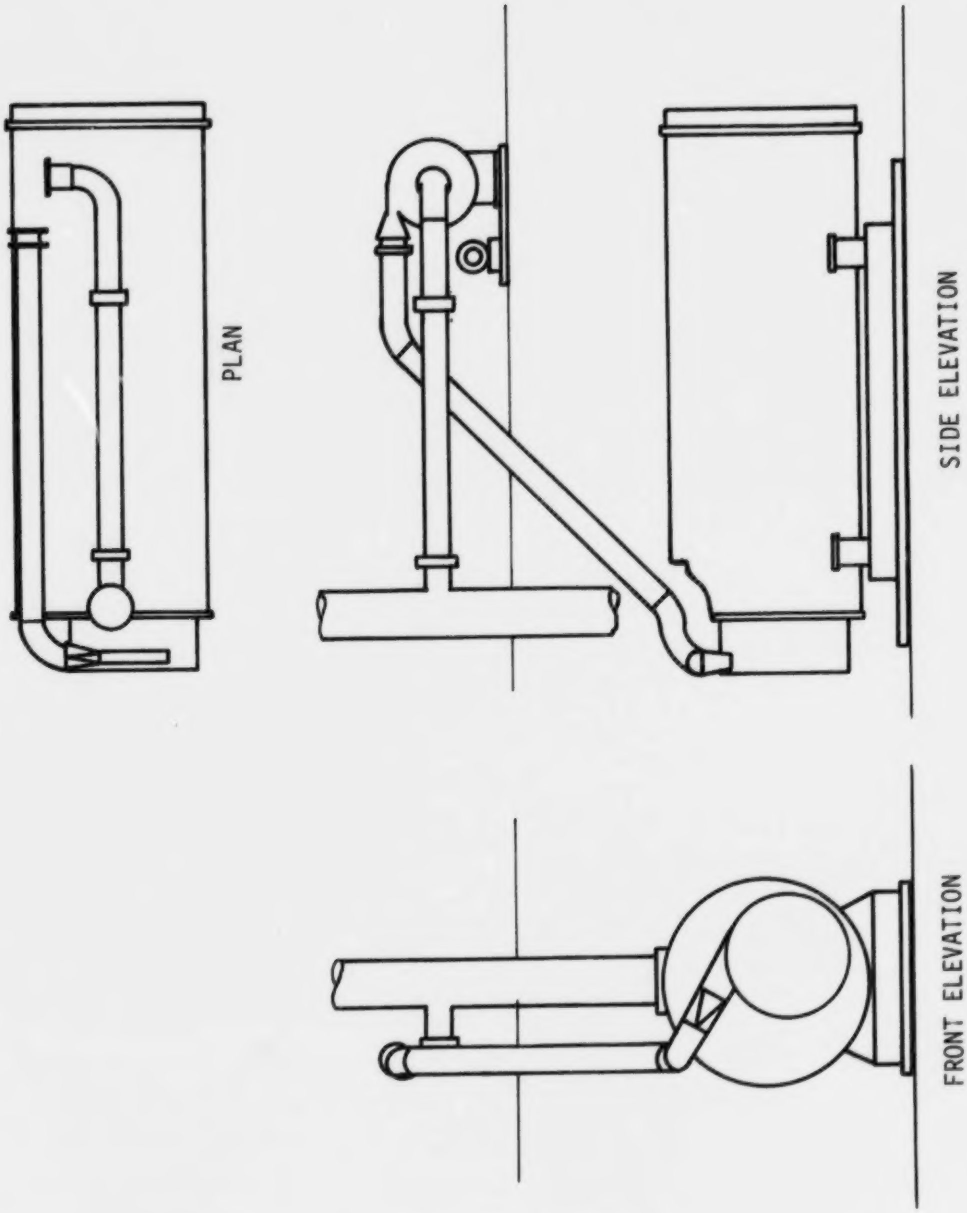

PLAN
SIDE ELEVATION
FRONT ELEVATION

Figure 5-8. Layout of Flue Gas Recirculation System for the Firetube Boiler.

prevent low frequency rumble.

The only practical entry for the second stage combustion air supply was through the rear of the firetube unit, although this was made difficult because the boiler backed up to a 3-ft thick wall leaving only limited space for access. Pentration of the front wall was rejected because it would necessitate cutting through 30 inches of refractoyr in the form of three cast refractory rings. The general arrangement of the staging system installed in the firetube boiler is shown in Figure 5-9. Ambient air was supplied by separate fan to a distribution ring at the rear of the boiler. This ring supplied eight 2-inch stainless steel staging pipes which entered the firetube through the rear door. These pipes were laid along the wall of the firetube and provision was made to allow the axial location of the staging injectors to be varied in later experiments. The staged air was injected radially through fishtail orifices. Burner stoichiometry was varied by throttling the combustion air supply and maintaining a constant overall excess air by increasing the air flow through the staging injectors.

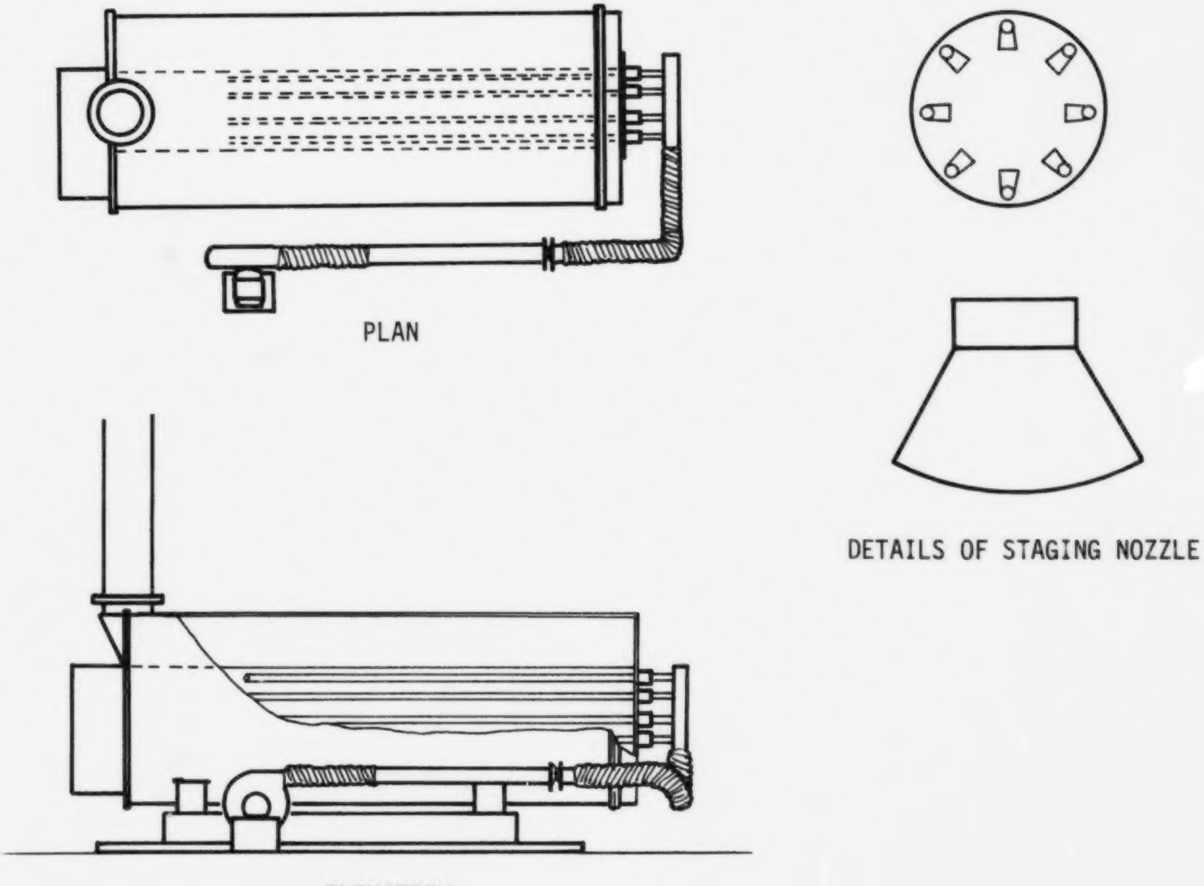
5.2.3 Fuel Delivery System for Methanol Evaluation

For the duration of the methanol evaluation tests, the methanol fuel was stored in a portable tanker adjacent to the boiler house. A fuel system was fabricated to deliver the methanol to the watertube and firetube. Copper tubing of 1-1/8 in. I.D. was used as the fuel supply line. A positive displacement rotary pump delivered the methanol to the watertube or firetube fuel nozzle at a pressure dependent on fuel flowrate.

The volumetric heating value of methanol is half that of fuel oil, thus if the fuel nozzles which are originally included in both boilers were not replaced, firing rates would be restricted to less than half of the maximum design value. To enable firing at the maximum level, a fuel nozzle of the same design, but larger flow capacity, was installed in the firetube boiler. Because of restrictions in methanol availability and limited steam requirements of the host site, the watertube boiler could never be fired at rates approaching its maximum capacity.

5.2.4 Flue Gas Measurement Systems - Field Investigations

The sample was withdrawn from a port in the flue of each boiler via a 0.5 in. O.D. x 0.049 in. wall stainless steel tube and passed to a condensate trap immersed in an ice bath. The cooled sample gas was supplied via a 1/4 in. O.D.



PLAN

ELEVATION

DETAILS OF STAGING NOZZLE

5-9. Details of Equipment Used in the Staging Investigation.

Teflon tube to the instrument manifold. The instruments used to continuously monitor the concentration of several combustion products are listed below; in some instances, backup instruments were employed to ensure continuity if one instrument failured during any particular test.

NO/NO$_x$. The primary instrument used to determine both NO and NO_x was a Thermo-Electron Chemiluminescence Analyzer, backup measurements were made with a Theta Sensor US-6000 analyzer.

O_2. Oxygen concentrations were measured by both the Theta Sensor analyzer and a Teledyne Model 320 AX Portable Analyzer with a Class A-3 cell, the former instrument being used as the primary reference.

CO. An infrared absorption analyzer (MSA Model LIRA-303) was used to determine the carbon monoxide content of the flue gases.

SO_2. The Theta Sensor US-600 was used to determine the sulfur dioxide content of the flue gases.

Smoke. Smoke readings were taken with a Bacharach smoke tester.

Certified zero and calibration gases were used to calibrate these monitors throughout the investigation. As with the laboratory tests, calibration checks were frequent.

The instrumentation available in the boiler house was used to establish the boiler load. Although this was adequate for normal operation, more precise information of fuel and steam flow would have been desirable for this investigation.

6.0 BOILER MODIFICATIONS FOR NO_x CONTROL

Under EPA Contract 68-02-0222(1) the effectiveness of flue gas recirculation as an NO_x control technique was investigated. During the present investigation further information was obtained with the same system to augment previous data. In the previous study the information obtained was limitted to oil firing and the system was not optimized. The investigations with the staged air injection system were extended to include an investigation of the influence of injector location and fuel type.

6.1 Flue Gas Recirculation

The equipment modification to allow the recirculation of cooled combustion products to the windbox of both the firetube and watertube boilers has been described in Section 5. The flue gas recirculation investigations had two major objectives:

- to check the validity of earlier data obtained with No. 5 fuel oil; and
- to extend the data base to include information on the performance of the equipment when firing natural gas and methanol.

Figures 6-1 and 6-2 compare the effect of FGR on NO_x emissions from the firetube and watertube boilers for two test series. The results are presented as a function of the mass percentage of FGR defined by:

$$\frac{\text{Mass of flue gas recycled}}{(\text{mass of fuel}) + (\text{mass of combustion air}) + (\text{mass of recycled flue gas})}$$

The results obtained for the two test series with the firetube boiler are very similar. However, considerable differences can be seen between test series results with the watertube, particularly at the lower load. In the first test series emissions increased slightly as flue gas was added to the combustion air. This increase did not occur in the second test series. For both boilers (when firing oil in both test series), 30 percent reduction in emissions appears to be the maximum achievable with flue gas recirculation before flame stability becomes a problem.

As expected, flue gas recirculation was particularly effective in reducing the emission from both boilers when firing natural gas. The results presented in Figure 6-3 show reductions on the order of 70 percent with 25 percent

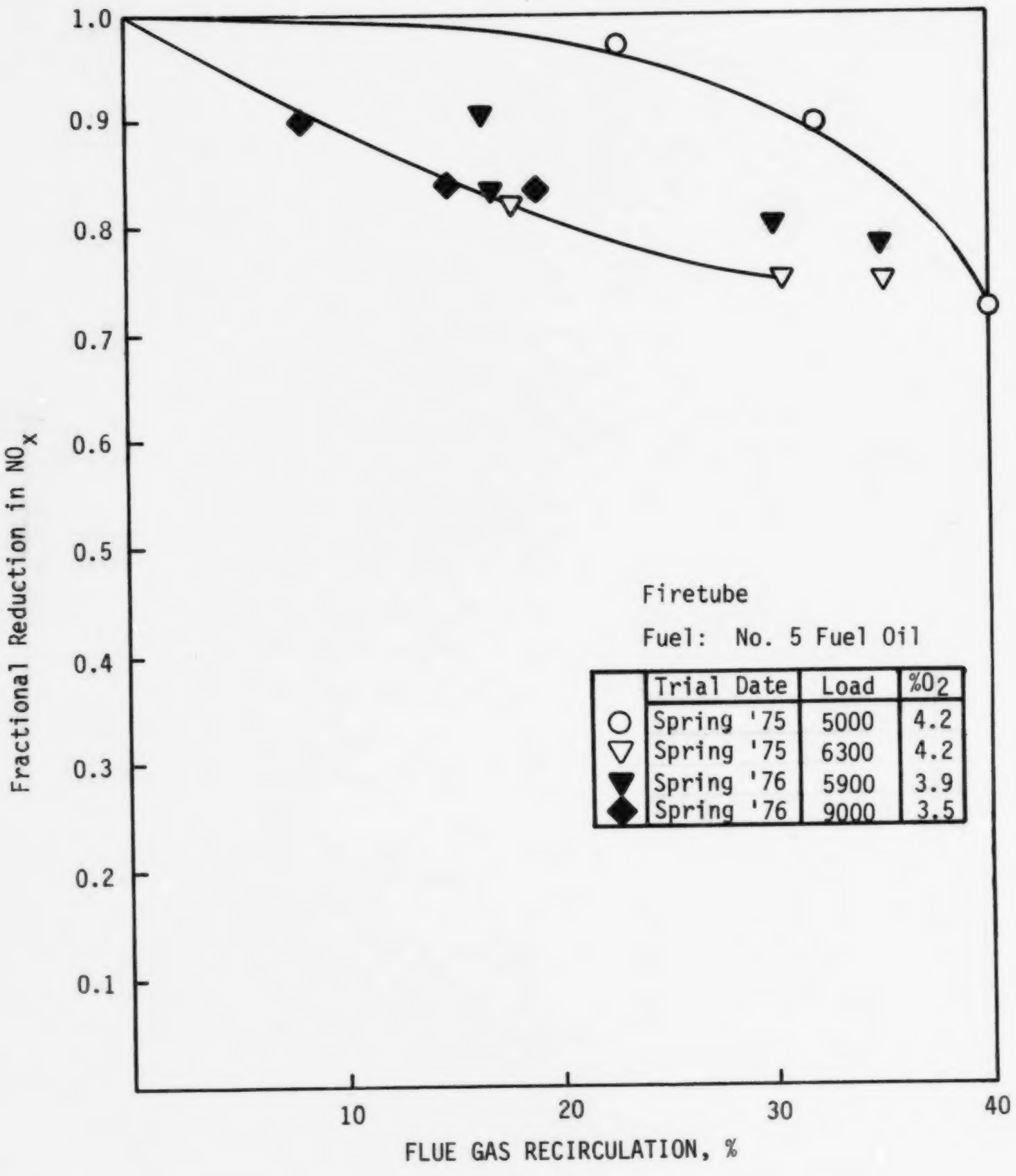


Figure 6-1. Fractional Reduction in NO$_x$ Achieved by Flue Gas Recirculation No. 5 Oil.

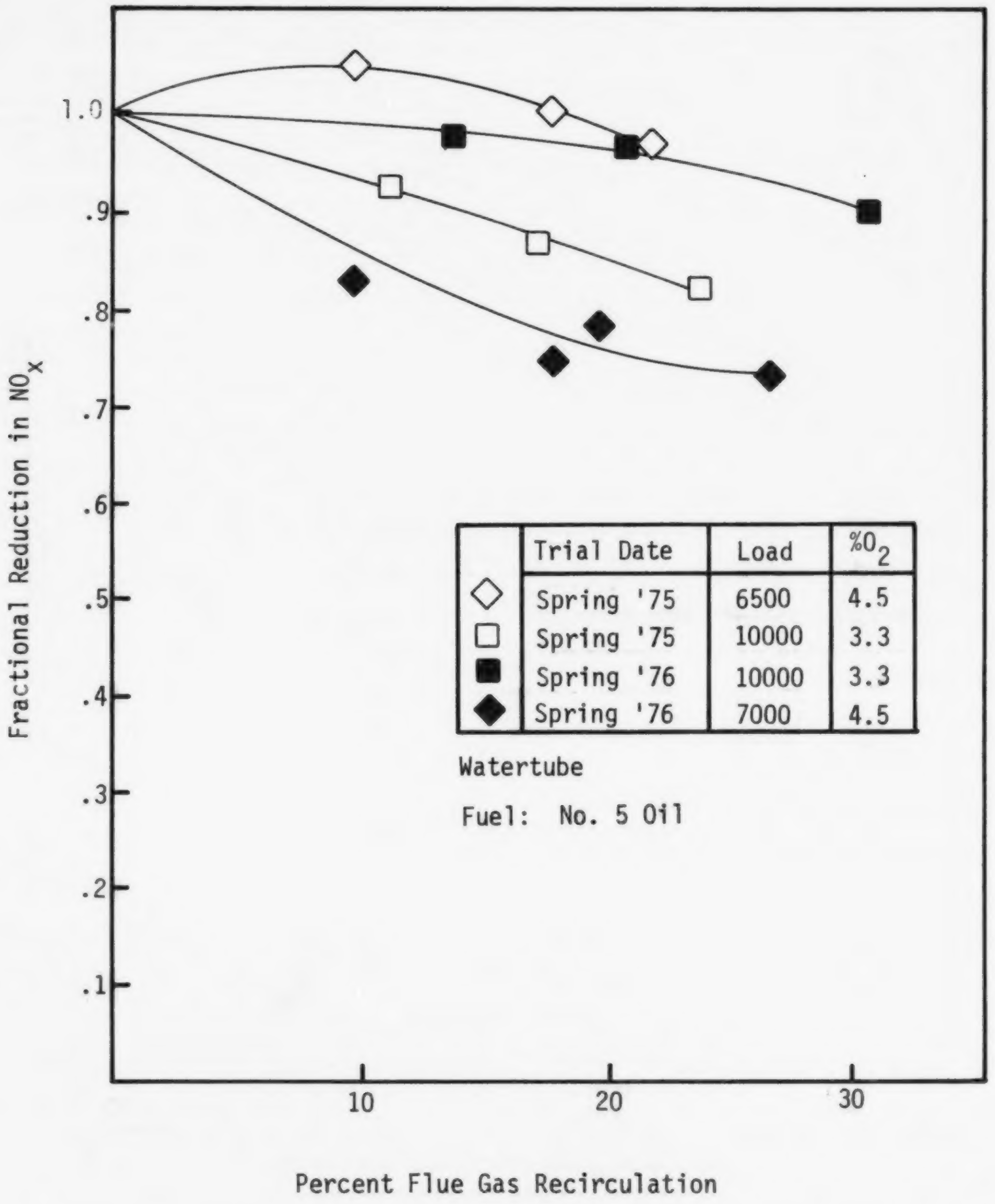


Figure 6-2. Fractional Reduction in NO_x Achieved by Flue Gas Recirculation with No. 5 Fuel Oil.

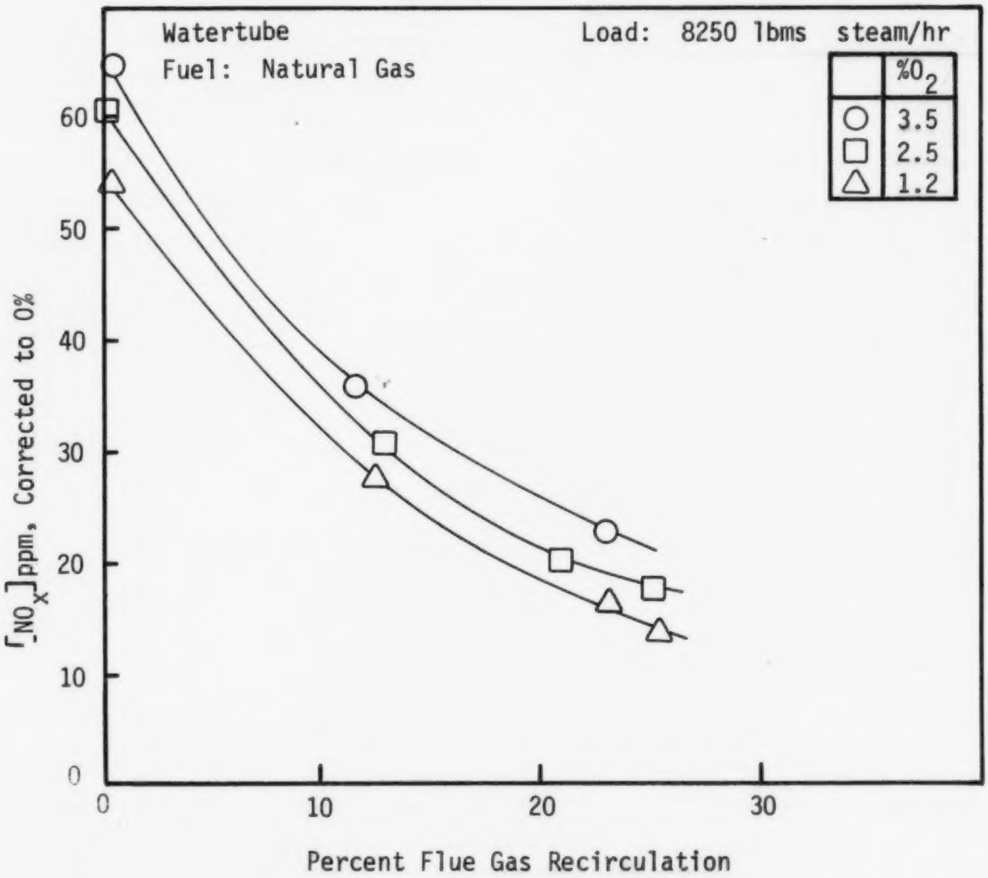


Figure 6-3. Percent FGR versus NO_x for Natural Gas Firing with the Watertube Boiler.

recirculation for the watertube boiler. The investigations were limited to this low load because of the small load demand during the period of natural gas availability. The influence of flue gas recirculation on emissions from the firetube boiler at two loads and various excess air levels is given in Figures 6-4 to 6-7. Excess air appears to have a minor effect upon NO_x levels with flue gas recirculation (see Figures 6-4 to 6-6), but a more significant effect upon carbon monoxide levels (see Figures 6-5 and 6-7). At intermediate loads the addition of flue gas recirculation to the combustion air appears to allow operation at reduced excess air levels with reduced CO and smoke emissions. This can be attributed to the improved mixing achieved by an increase in the momentum of the air stream. The relatively high smoke emissions at low excess air gives an indication of the normally low rate of fuel/air mixing when this particular boiler fires natural gas.

It has been reported earlier that pulsation problems were experienced in the firetube boiler when firing fuel oil and using FGR. Pulsations were not encountered with natural gas, suggesting that the original problems were associated with the oil feed system.

The effectiveness of flue gas recirculation with methanol firing is very similar to that reported with natural gas. Both boilers behaved similarly (see Figures 6-8 and 6-9). There does not appear to be any strong influence of load, excess air level or nozzle size on the effectiveness of FGR in either the firetube or the watertube boiler.

Figures 6-10 and 6-11 compare the influence of flue gas recirculation on NO_x emissions for the three fuels tested under similar conditions. The lowest levels were achieved with methanol, although the greatest absolute reduction was achieved with natural gas. Even though the No. 5 fuel oil only contained approximately 0.1 percent nitrogen, it can be seen that flue gas recirculation has a relatively small effect on NO_x emissions compared to fuels which do not contain bound nitrogen. Also, flue gas recirculation with natural gas firing can give emissions comparable to those achieved with straight methanol firing.

The results obtained from the field test concerning the relative effectiveness of FGR were different than those obtained with the laboratory firetube simulator. Figure 6-12 presents results concerning the effectiveness of flue gas recirculation for a number of fuels. The effects of FGR on NO_x emissions

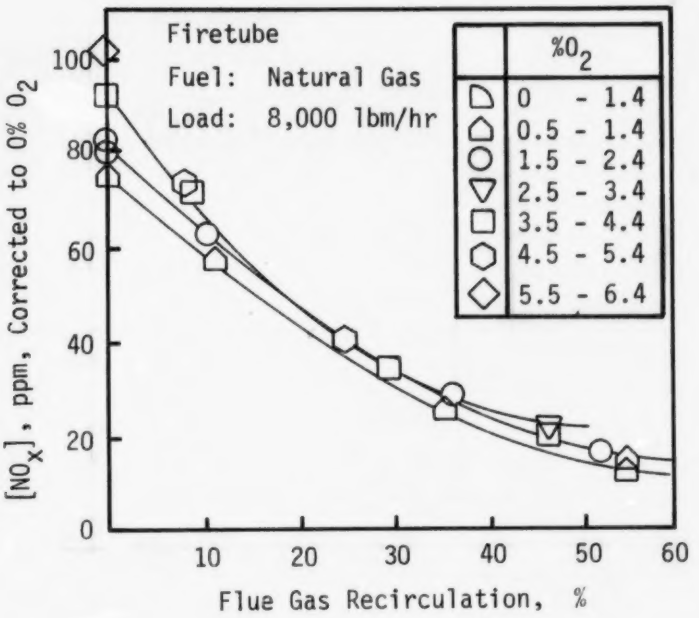


Figure 6-4. Percent FGR versus NO_x for the Firetube, Natural Gas.

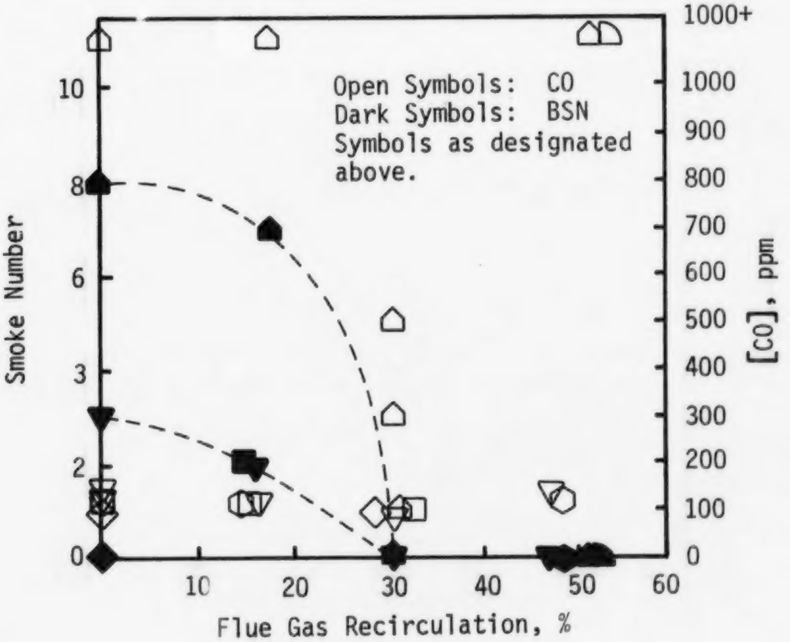


Figure 6-5. Percent FGR versus CO and Smoke for the Firetube, Natural Gas.

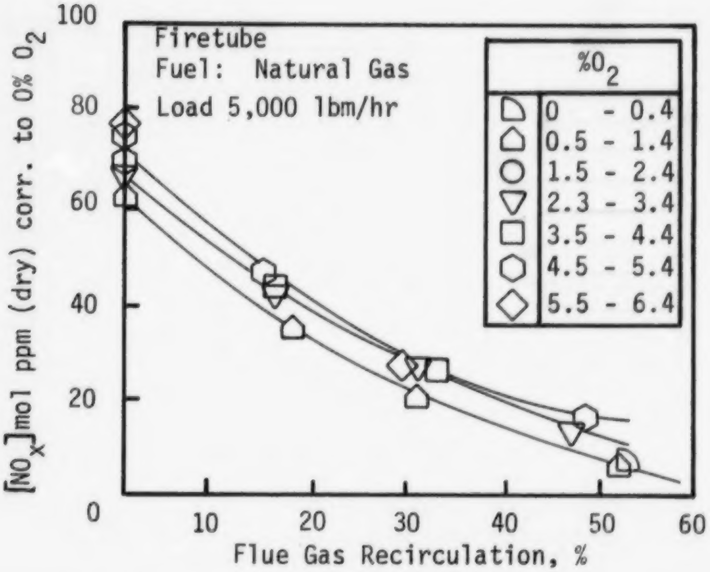


Figure 6-6. Percent FGR versus NO_x for the Firetube with Natural Gas Firing.

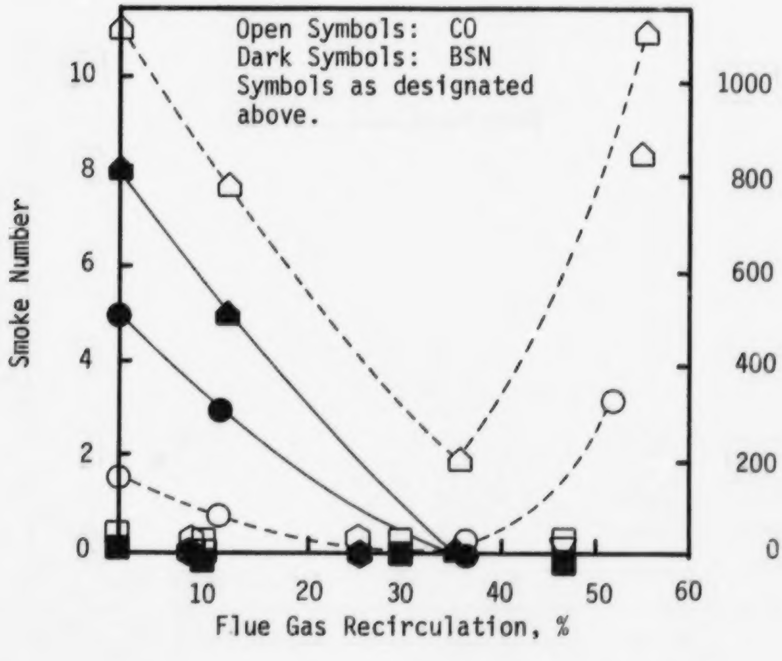


Figure 6-7. Percent FGR versus CO and Smoke for the Firetube.

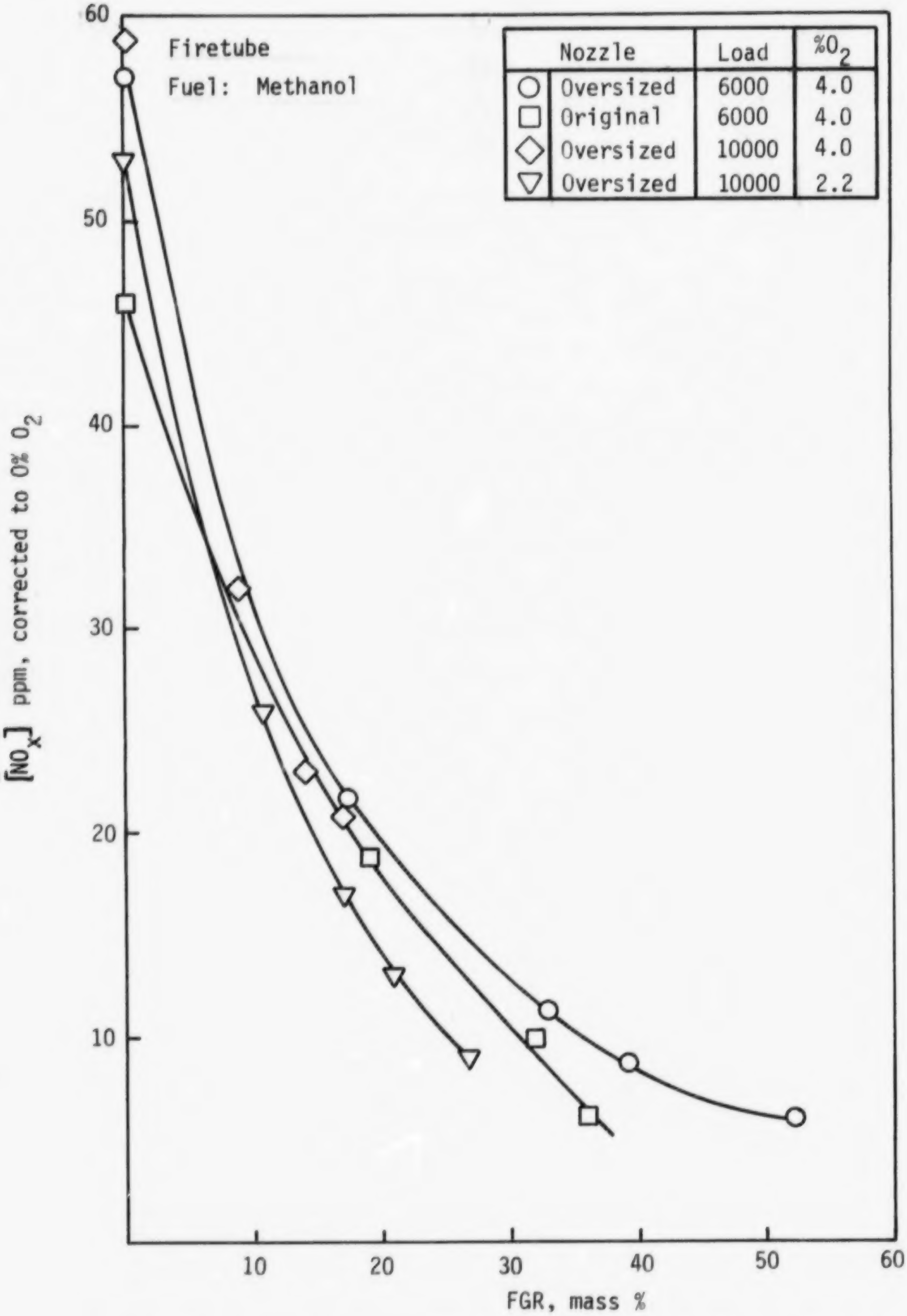


Figure 6-8. Effect of FGR on NO_x from Firetube Boiler with Methanol Firing.

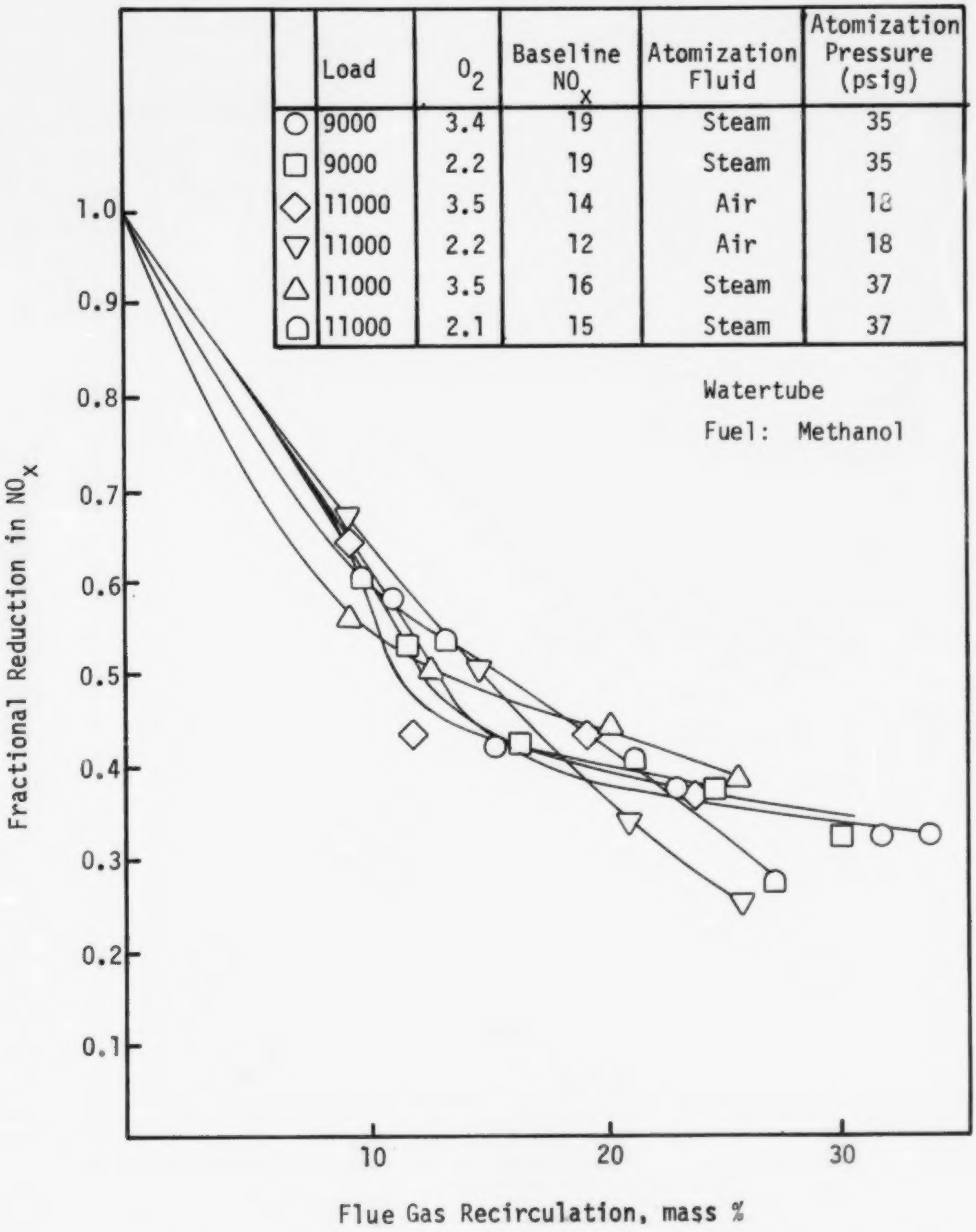


Figure 6-9. Effect of FGR on NO_x from Watertube Boiler with Methanol Firing.

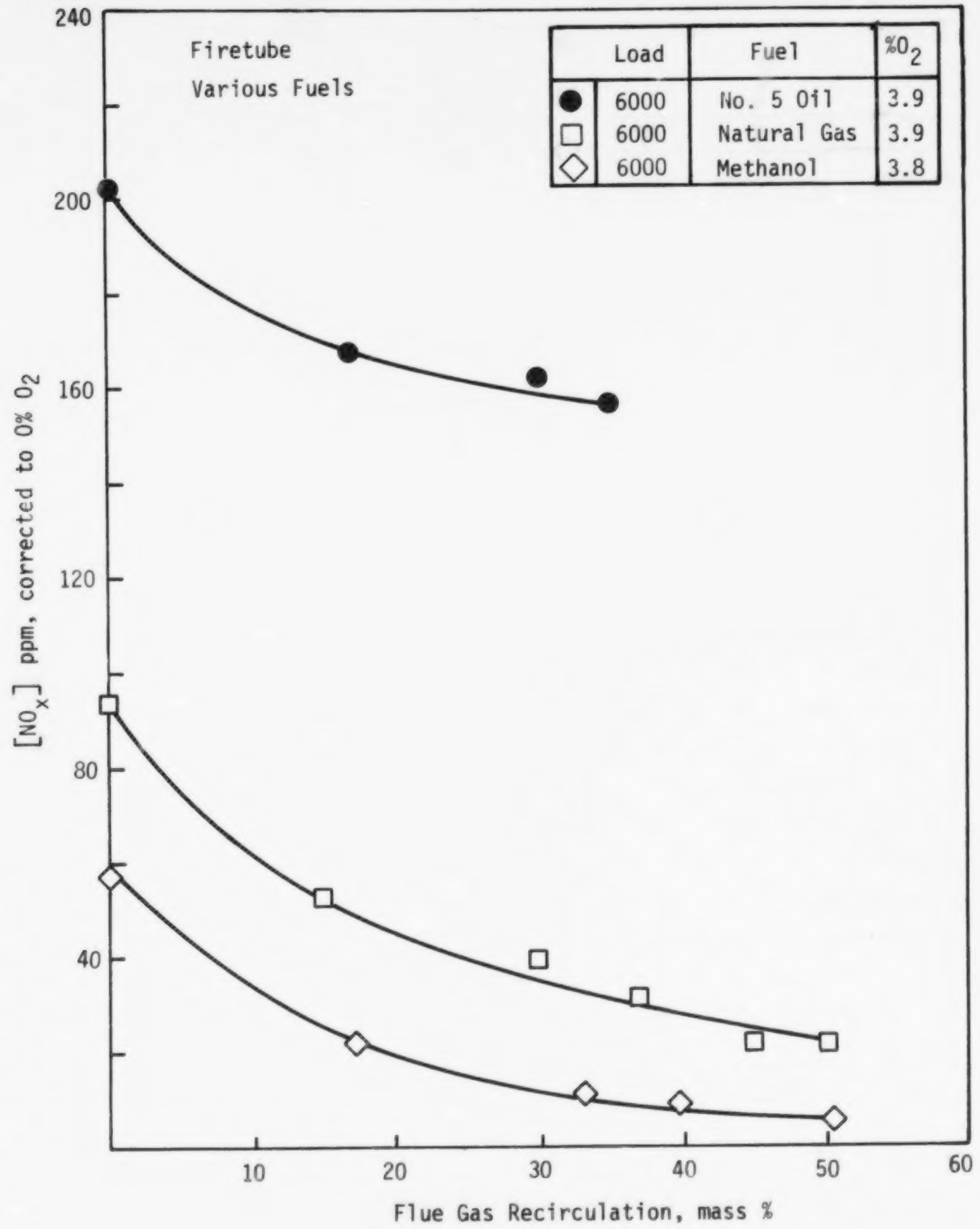


Figure 6-10. Comparison Of The Influence Of Flue Gas Recirculation on NO_x Emissions for Various Fuels.

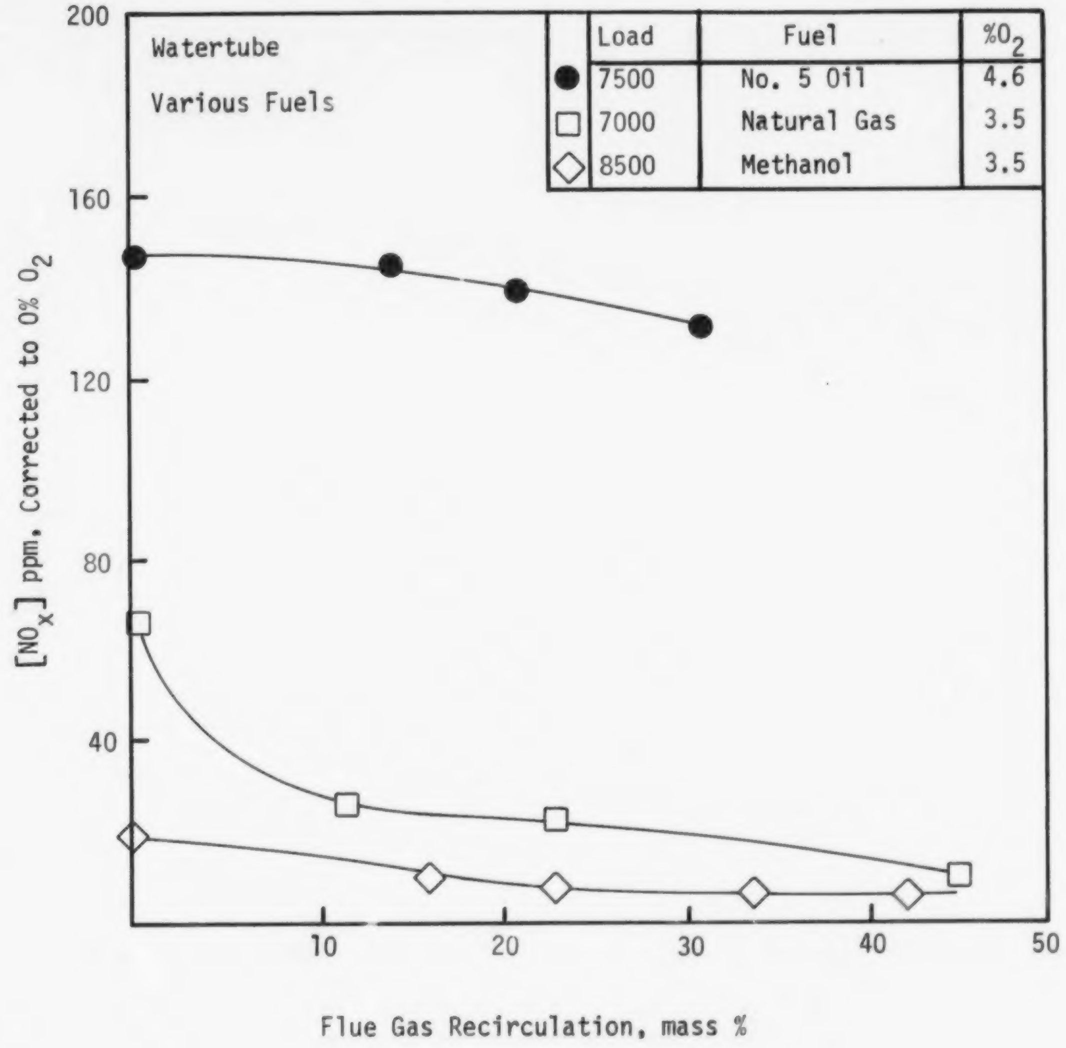


Figure 6-11. Comparison Of The Influence Of Flue Gas Recirculation on NO$_x$ Emissions for Various Fuels.

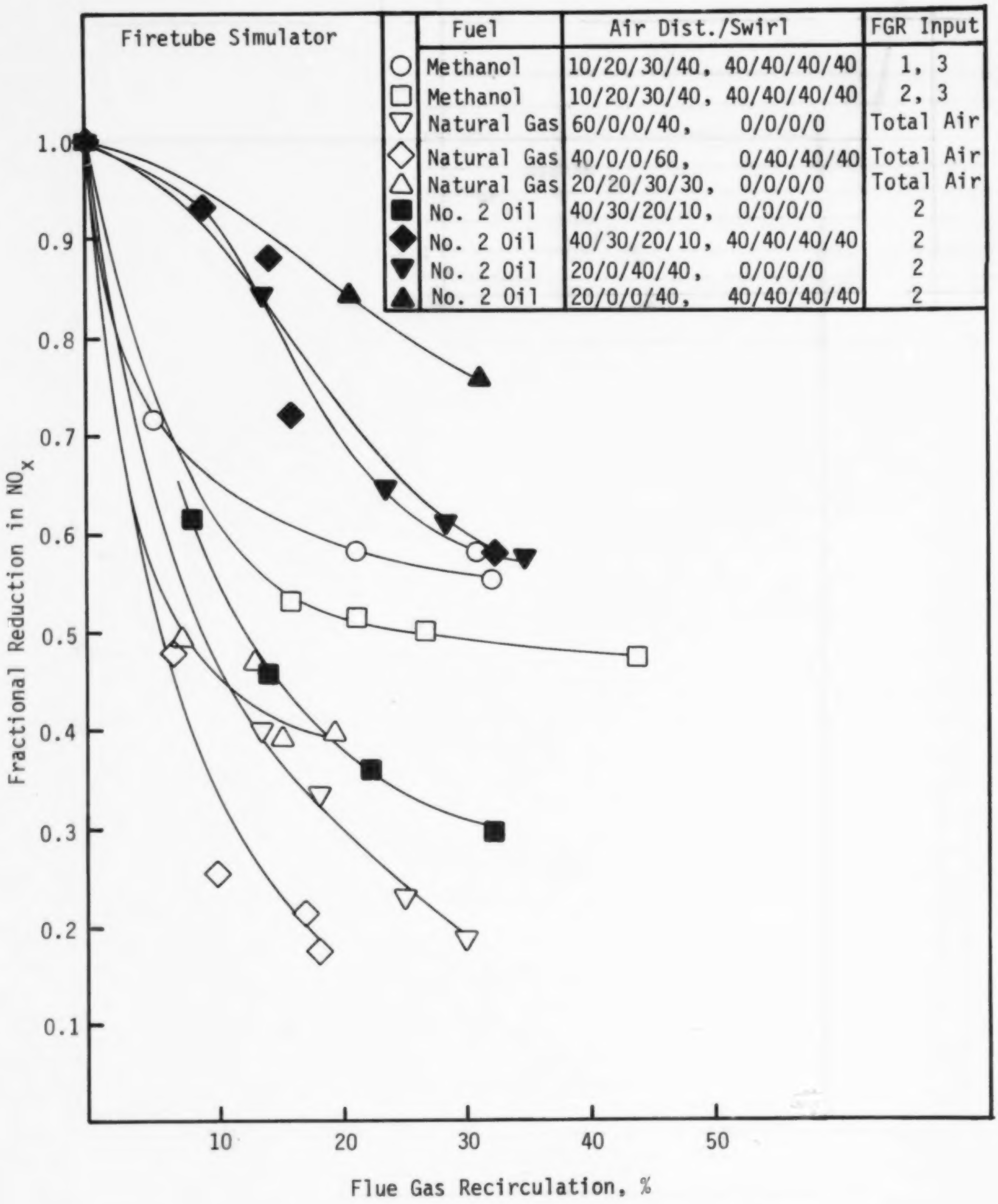


Figure 6-12. Effect of FGR on NO$_x$ from the Laboratory Firetube with Various Fuels.

from No. 2 oil is seen to be quite varied and dependent on the air and swirl configuration of the combustion air and location of the flue gas addition. For the natural gas and methanol, the effectiveness of flue gas recirculation appears to be much more dependent upon fuel type than upon combustion air distribution or upon the method of introduction of the flue gas. Figure 6-13 compares the effectiveness of flue gas recirculation obtained with methanol firing in the laboratory combustor and the two field operating boilers. It can be seen that the effectiveness differs in each system. The differences are doubtlessly attributable to variations in fuel/air mixing patterns.

6.2 Staged Combustion

In the investigations reported earlier(1) preliminary results were obtained for one fuel (No. 5 oil) and one staging ring location. It was found that the exhaust NO_x level was essentially independent of the overall excess air, but as expected, strongly dependent upon burner stoichiometry. The influence of burner stoichiometry on NO_x emissions with No. 5 fuel oil firing is given in Figure 6-14 as a function of staging ring location. For any given airflow distribution the NO emissions are lowest with the staging ring farther from the burner, although this difference is less apparent at high staging rates. Figure 6-15 presents a cross-plot of the same data showing the effect of staging ring location on NO and CO emissions at constant burner stoichiometry. Smoke emissions did not increase, but CO emissions doubled with a 50 percent reduction in NO. The results presented in Figure 6-14 indicate that there appears to be an optimum burner stoichiometry which varies with the location of staged air addition. Further increases in staging percentage past this optimum stoichiometry cause an increase in NO_x, especially at the 1.5D and 2.5D staging ring locations. This observation may be interpreted in terms of the interaction between the percentage of staging and the intensity of fuel/air mixing at the burner exit. The major criteria for staging effectiveness include the overall equivalence ratio at the burner exit, the fuel/air mixedness in the staging section, and the residence time in the staging section. For a constant load and overall excess air level, increases in the percentage of combustion air staged decrease the airflow at the burner exit which may reduce the intensity of the fuel/air mixing process. It is possible that after a certain burner stoichiometry is reached further increases in staging percentage decrease fuel/air mixing rates sufficiently to

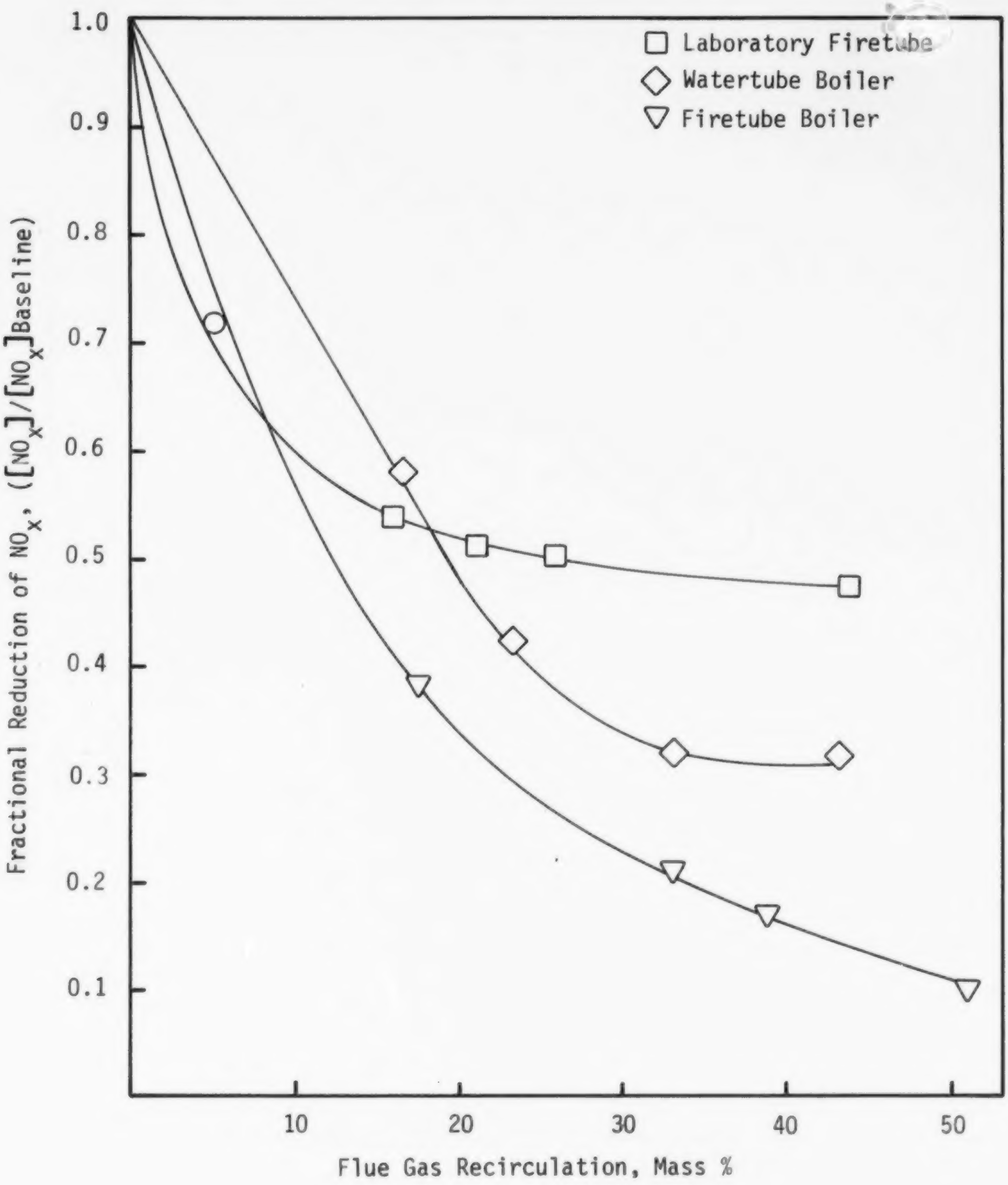


Figure 6-13. FGR Effectiveness with Methanol Firing for the Field Boilers and the Laboratory Combustor.

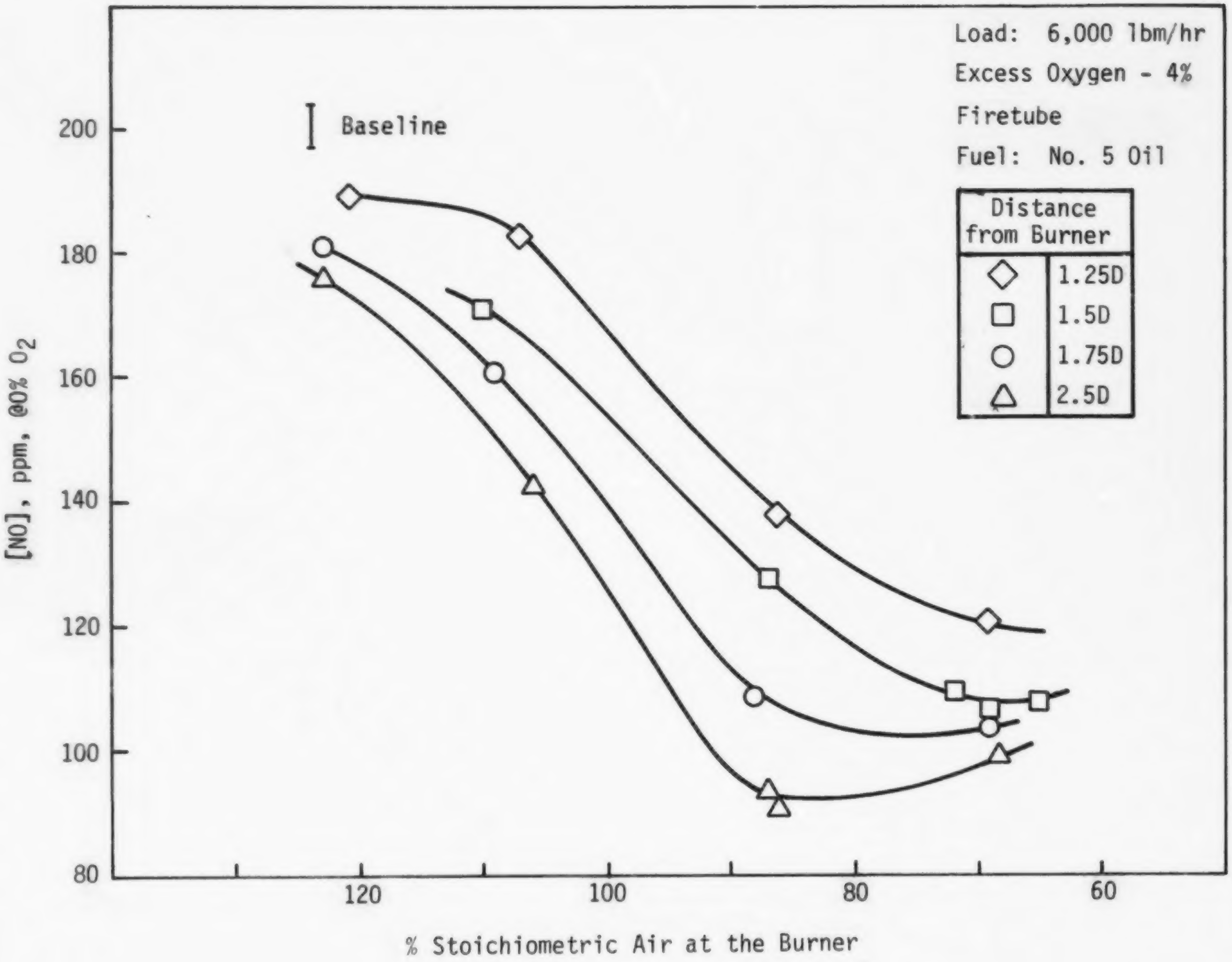


Figure 6-14. The Influence of Staging Ring Location on NO_x Emissions from the Firetube with No. 5 Fuel Oil.

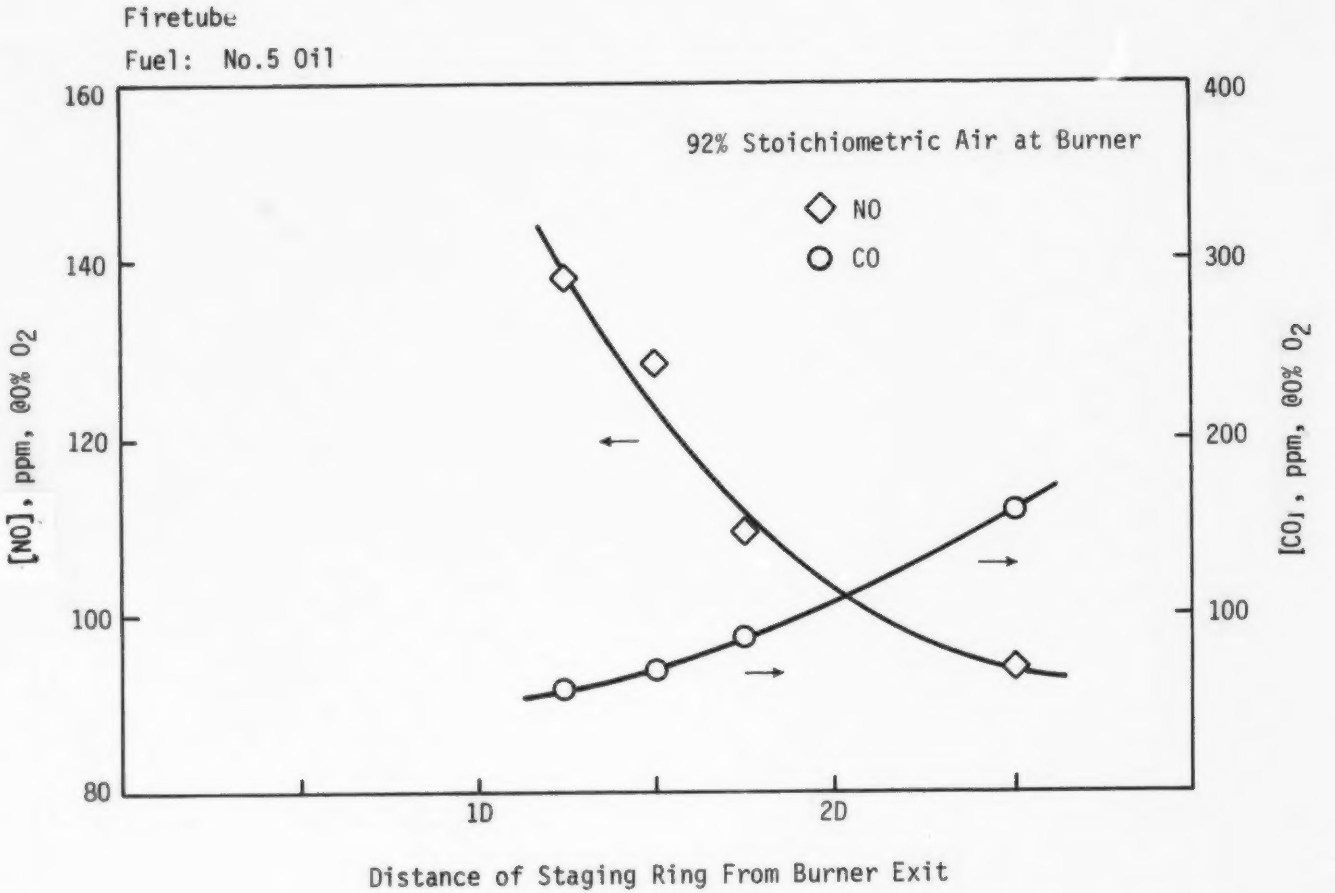


Figure 6-15. Effect of Staging Ring Location on NO Emissions from the Firetube with No. 5 Fuel Oil Firing.

inhibit the staging effectiveness. It should be noted that baseline emissions were effected by presence of the staging ring.

The results for natural gas firing with the same system are presented in Figures 6-16 to 6-18. Figure 6-16 shows the effect of burner stoichiometry on NO emissions for two excess air levels at constant load and staging ring location. It can be observed that there is an optimum staging percentage for a given load, staging ring location and overall excess air level. For fixed staging ring location, the minimum NO level occurs at a lower percentage of staging for the higher overall excess air level, and is marginally higher in magnitude than the minimum observed at the lower excess air level. The difference in location of the NO minimum could be due to the higher air exit velocities for the higher excess air levels.

Figures 6-17 and 6-18 present the same data as that presented in Figure 6-16 replotted as a function of staging percentage for several staging ring injector locations. The effect of staging ring location is more apparent, and it can be seen that the lowest emissions were not obtained with the ring furthest from the burner exit nor were the highest emissions obtained with the ring closest to the burner. This could be due to the type of interaction previously described between burner exit velocity (fuel/air mixing), burner equivalence ratio, and staging residence time. Additionally, the presence of the staging ring could effect combustion air mixing in the secondary oxidation region and thus NO formation.

With fuels containing bound nitrogen the reduction in NO_x emissions obtained by staging the combustion process can be explained by the reduced oxygen availability in the early stages of heat release. Fundamental studies have shown that conversion of fuel nitrogen compounds to nitrogen oxides in flames is strongly influenced by the reactant stoichiometry and that the percentage conversion decreases rapidly as the percentage of theoretical air supplied with the fuel falls below 100 percent. With non-nitrogen containing fuels staging can have two effects:

1. It reduces peak temperatures - ideally the total combustion process is divided into two stages and provided heat is removed between the stages, peak temperatures in the second stage will be lower than those that would have been produced in a single stage process without heat release.

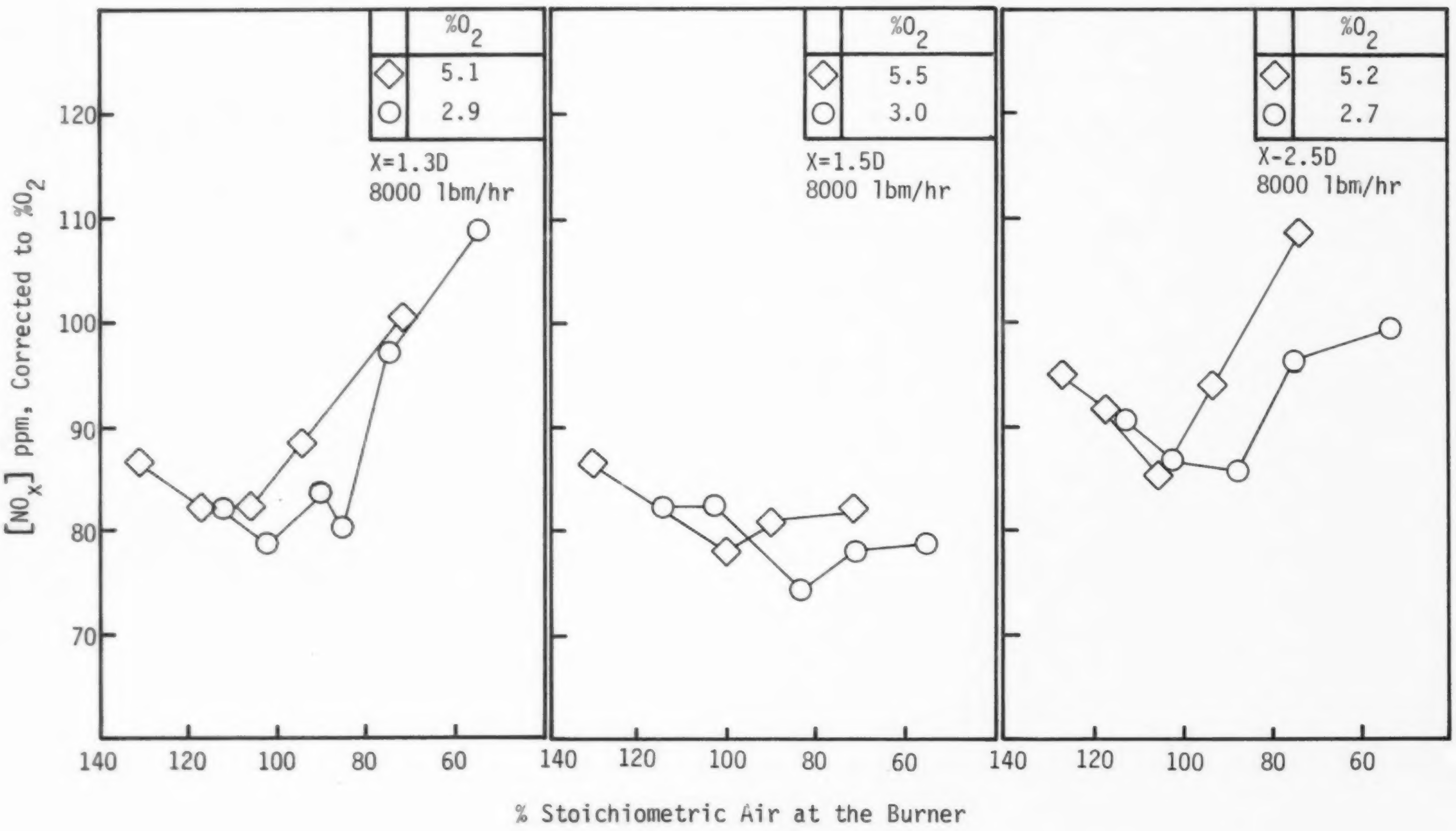


Figure 6-16. Effect of Burner Stoichiometry on NO_x Emissions as a Function of Staging Injector Location and Overall Excess Oxygen.

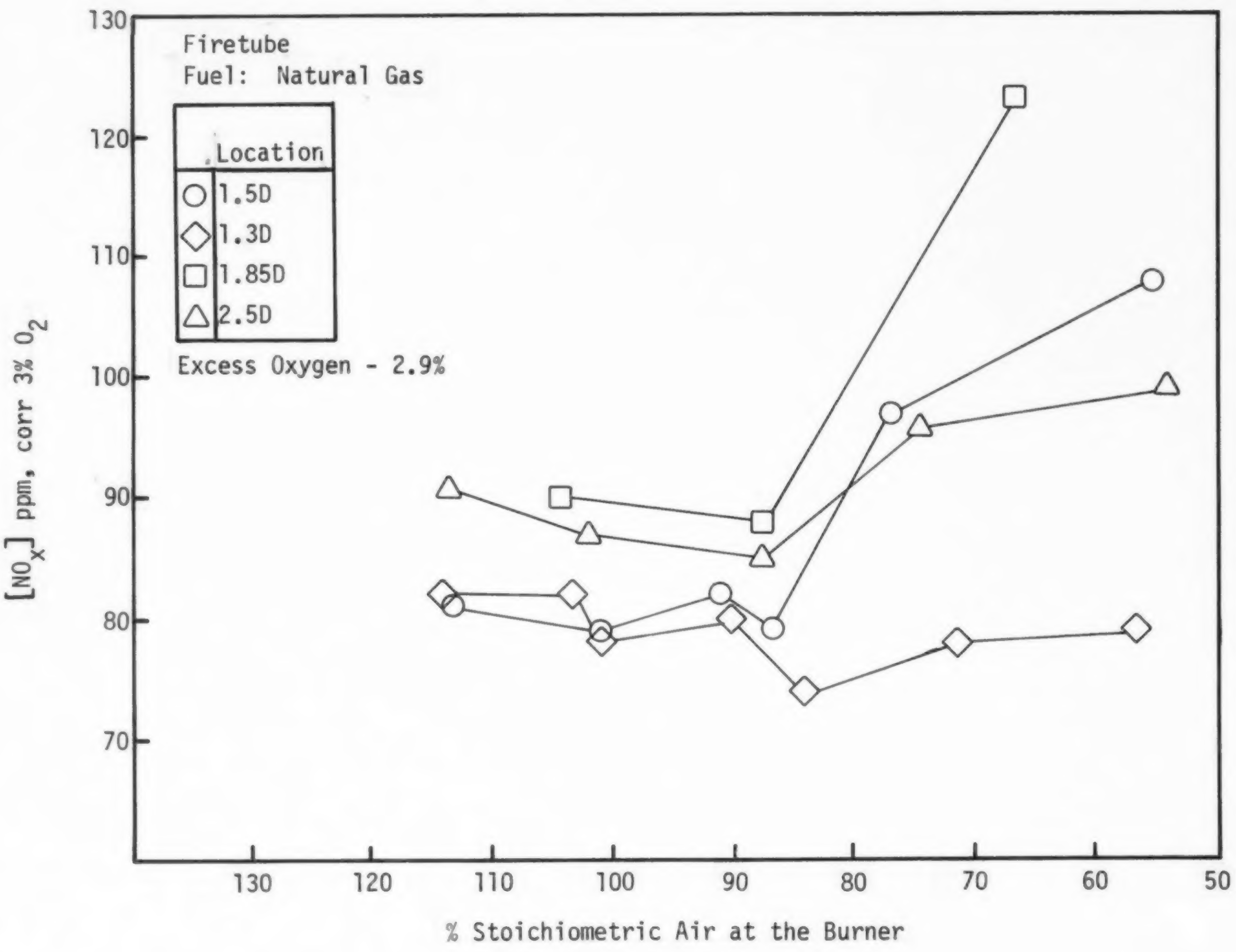


Figure 6-17. Influence of Location of Staged Air Injection (2.9 Percent Oxygen).

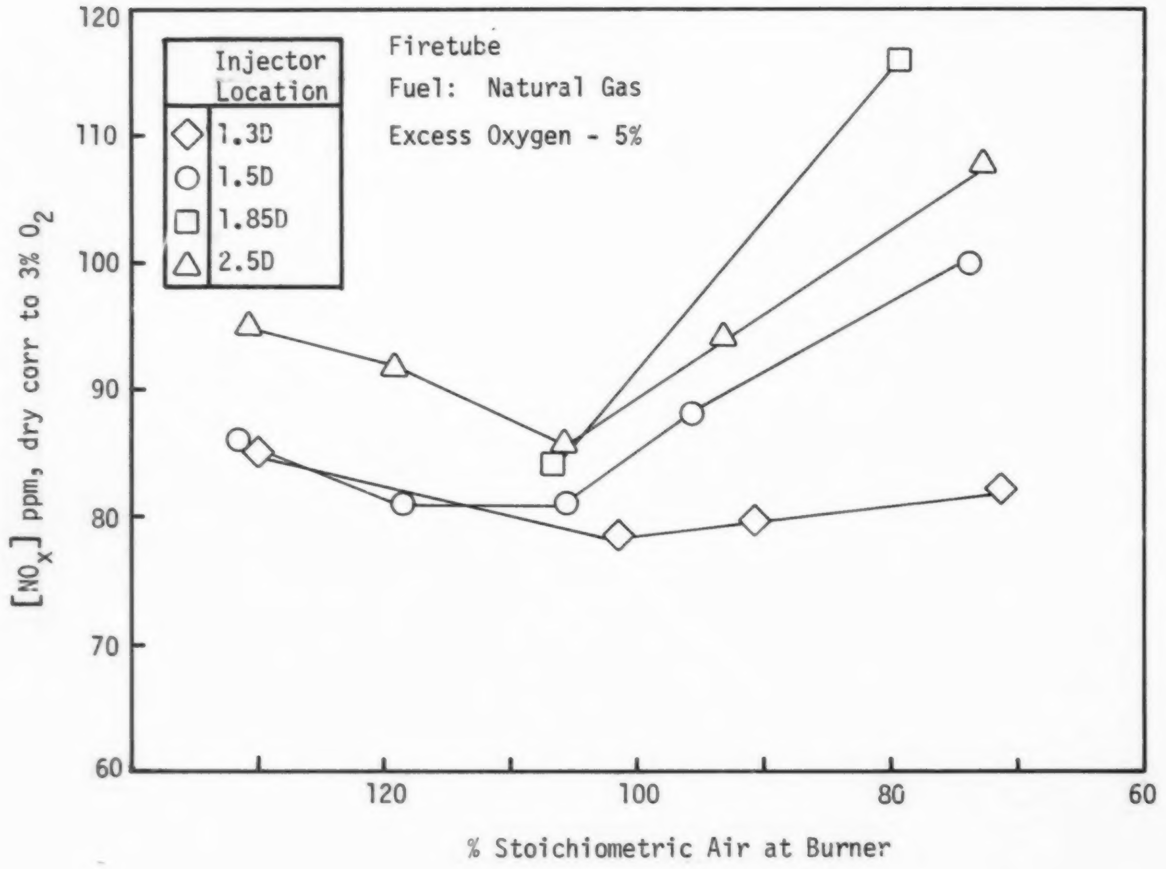


Figure 6-18. Influence of Location of Staged Air Injection (5 Percent Oxygen).

2. Under rich conditions hydrocarbon fragments attack nitrogen molecules forming nitrogen-containing species that are readily converted to nitric oxide under oxidative conditions. Although these same reactions would occur in oil flames, the nitrogen species produced from the fuel would tend to reduce their significance.

It should be remembered that mixing and reaction occur simultaneously in the practical system with heat transfer; therefore, staged heat release/removal occurs naturally in practice. Under staged conditions the fuel is mixed with part of the air and reacted in the first stage. The balance of the combustion air is injected into the combustor downstream of the fuel injector. Thus the transfer processes in the first stage dictates the state of the reactants entering the second stage (i.e., whether all fuel is reacted to CO + H_2 or whether part survives unreacted).

Two main features differentiate the staging results obtained with natural gas from those reported earlier in this section for fuel oil:

- Emissions do not decrease monotonically as the point of the staged air injection was located at increasing distances from the fuel injector. Emissions show both a maximum and a minimum which appear to be exaggerated as the first stage becomes more fuel-rich. This observation is illustrated by the results shown in Figures 6-19 and 6-20.
- As shown in Figure 6-16, NO_x emissions are not independent of overall excess air for fuel-rich first-stage conditions. This is particularly apparent for the air injector location farthest from the fuel injector.

The idealized picture of staged combustion effects discussed above does not adequately describe the mixing processes that must occur in practical systems which could account for the unexpected trends. However, the results obtained in the firetube boiler do not represent the first attempt to stage natural gas combustion which has resulted in an increase in NO_x emission.(23)

One reasonable suggestion which could account for the observation is based upon the concept that NO formation in gaseous turbulent diffusion flames is strongly dependent upon the rate of entrainment of bulk gases into the reacting mixture since this will control the residence time of the combustion products.

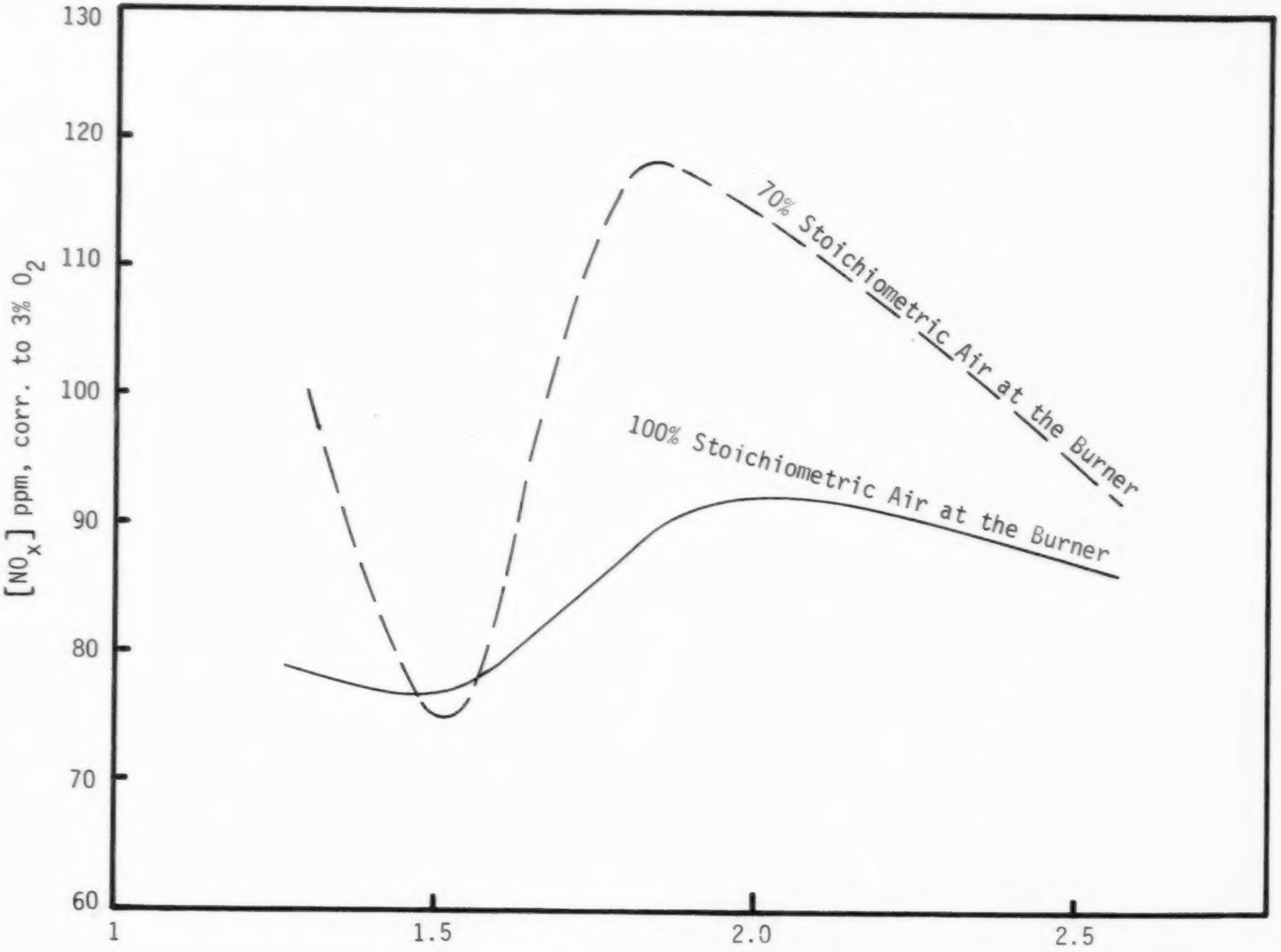


Figure 6-19. Injector Location in Combustor Diameters (2.9% Oxygen).

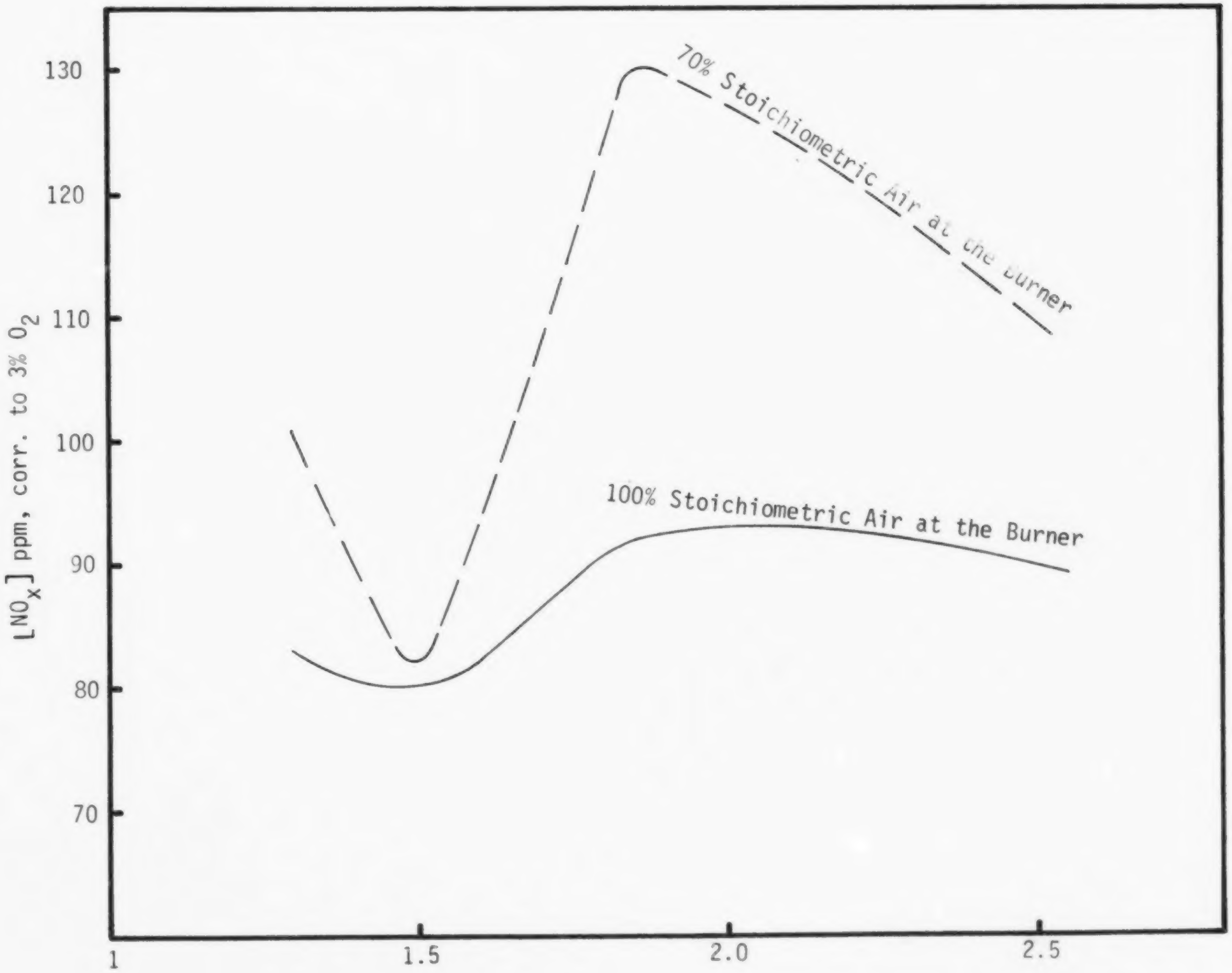


Figure 6-20. Injector Location in Combustor Diameters (5% Oxygen).

Since reaction zone temperatures are quenched by mixing with the cooler bulk gases the more vigorour the mixing process the more rapid is the quench rate and the resulting residence time at high temperature is reduced. This mixing process is controlled to a large extent by the burner momentum which is reduced as the percentage of stoichiometric air at the burner is reduced. The mixing process occurring within the second stage is also important and acts in a similar manner. Consider that if the first stage were to be perfectly mixed, and at the location of staged air injection the first stage mixture formed a uniform plug of partially oxidized fuel, then the second stage air addition could be considered as two extremes:

1. the second stage air is added and mixes instantaneously producing fully combusted products whose temperature is adiabatic for the reactants;
2. the second stage air is added in finite successive amounts which also mix instantaneously and give adiabatic products. However, the mixture now progressively becomes less rich, reaches stoichiometric, and then becomes fuel-lean. If no heat has been lost, the final temperature will be the same as in 1. above, but at some time when the reactants were mixed in stoichiometric proportions, maximum temperatures would occur far in excess of those achieved in 1.

Thus these two extremes represent conditions that would produce high and low NO levels. The real process must be somewhere between these extremes, but they illustrate how the mixing processes that occur in the second stage could influence the final NO level. Heat loss will occur simultaneously with these mixing effects and will be proportional to the heat released (and therefore the primary stoichiometry). Also, the enthalpy of the first stage products will depend upon the heat loss, and therefore as the staged air injector is moved, mixing and heat loss effects could act in opposition and cause the minimum and maximum shown in Figures 6-19 and 6-20.

An alternative hypothesis could be proposed which could also contribute to the observed effects which depends upon oxidation of nitrogen species in the second stage. Under fuel-rich conditions it is known that reactions such as

$$CH + N_2 \rightarrow CHN + N$$

and

$$CH_2 + N_2 \rightarrow CHN + NH$$

Subsequent reactions then allow the production of either NO or N_2 and the yield of NO would depend upon temperature, hydrocarbon fragment concentration and oxygen concentration. A plausible mechanism of HCN oxidation involves the production of an intermediate radical NCO

$$HCN + H \rightarrow CN + H_2$$

$$HCN + OH \rightarrow CN + H_2O$$

$$CN + OH \rightarrow NCO + H$$

$$H + NCO \rightarrow NH + CO$$

A kinetic path then exists for the production of NO or N_2 via

$$NH_x + NO \rightarrow N_2 + H_2O$$

or

$$NH_x + O_2 \rightarrow NO + OH_x$$

Temperature in the primary stage could be important since it will dictate the rate at which the nitrogen species (HCN, CN, NCO and NH_x) decay. Should second stage air be added before they are essentially zero, then they will be oxidized to NO and this will cause an increase in the final emission levels.

Consider the results shown in Figures 6-19 and 6-20 for 70 percent stoichiometric air at the burner. As the staged air injector is moved downstream, two different effects will occur. Close to the burner the fuel molecules will not "recognize" that staging is taking place since mixing is progressive, but as the distance increases and the time lengthens, the rich primary mixture will become more uniformly mixed allowing the production of nitrogen species with a finite lifetime. Emission will increase until the air is added at a time when the concentration of these species has begun to decrease and the final emission level will also decrease.

Both of the possible explanations given to account for the increase in NO_x emissions when natural gas flames were staged in the firetube boiler are speculative, but they are able to account for the observations.

7.0 EMISSIONS CHARACTERISITCS AS A FUNCTION OF BURNER DESIGN AND FUEL TYPE

The objectives of the investigations described in this section were not only to determine NO_x emissions from methanol and to compare them with emissions from conventional fuels, but also to assess practical problems associated with the utilization of methanol. The laboratory investigations were directed towards determining optimum conditions for methanol combustion and thus increasing the efficiency of the field tests. Naturally it was not possible to assess the interchangeability of methanol on all commercial burner types, but experience with four different burners strongly suggests that no insurmountable practical problems exist which would prevent the use of methanol in industrial boilers.

7.1 Laboratory Investigation

The laboratory firetube simulator, described in Section 5.1, was used to assess the NO_x emissions characteristics, effectiveness of pollutant control techniques, and boiler performance parameters for methanol in relation to other fuels. The emissions characteristics of methanol were determined for two different burner designs: 1) the MCSG was used to examine the effect of various fuel/air mixing patterns on NO formation, and 2) the commercial burner was used to evaluate the effect of changes in standard operating parameters (firing rate, excess air, etc.) on NO_x emissions.

7.1.1 Multi-Channel Swirl Generator (MCSG) Investigations

The MCSG was used with the divergent exit (Figure 5-3) to compare NO emissions from methanol and natural gas. The same axial and tangential velocity distributions were used with both fuels, and the results are presented in Figures 7-1 and 7-2. Data is presented as measured (dry sample, at 3% O_2) for several swirl configurations as a function of the degrees of rotation, X, of the swirl nozzle (or nozzles).

For the higher exit velocity case (50/0/0/50 air distribution, Figure 7-1) NO_x emissions from methanol were found to be less than half of those from natural gas. NO_x emissions from methanol appear to be relatively insensitive to swirl nozzle position, unlike the results presented for natural gas. The lower exit velocity case (40/0/30/30 air distributin, Figure 7-2) shows that under some swirl conditions, NO_x from natural gas is similar to methanol. The particular condition is that of low swirl on the inside channel, directly adjacent to the fuel nozzle with high swirl on the outside (Nos. 3 and 4) channels. A minimum

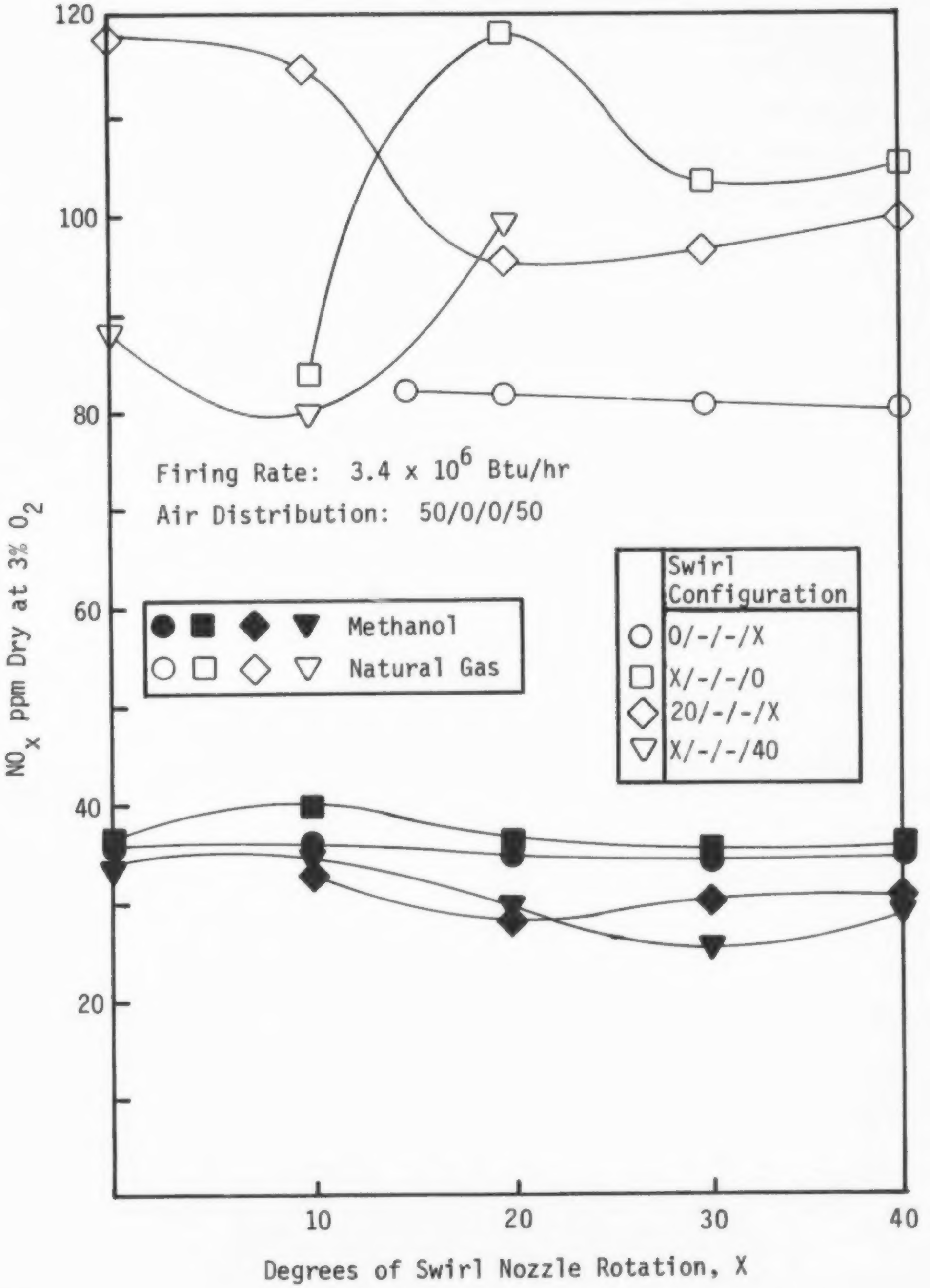


Figure 7-1. The Effect of Swirl Nozzle Rotation on NO$_x$ Emissions with Natural Gas and Methanol for a 50/0/0/50 Air Distribution.

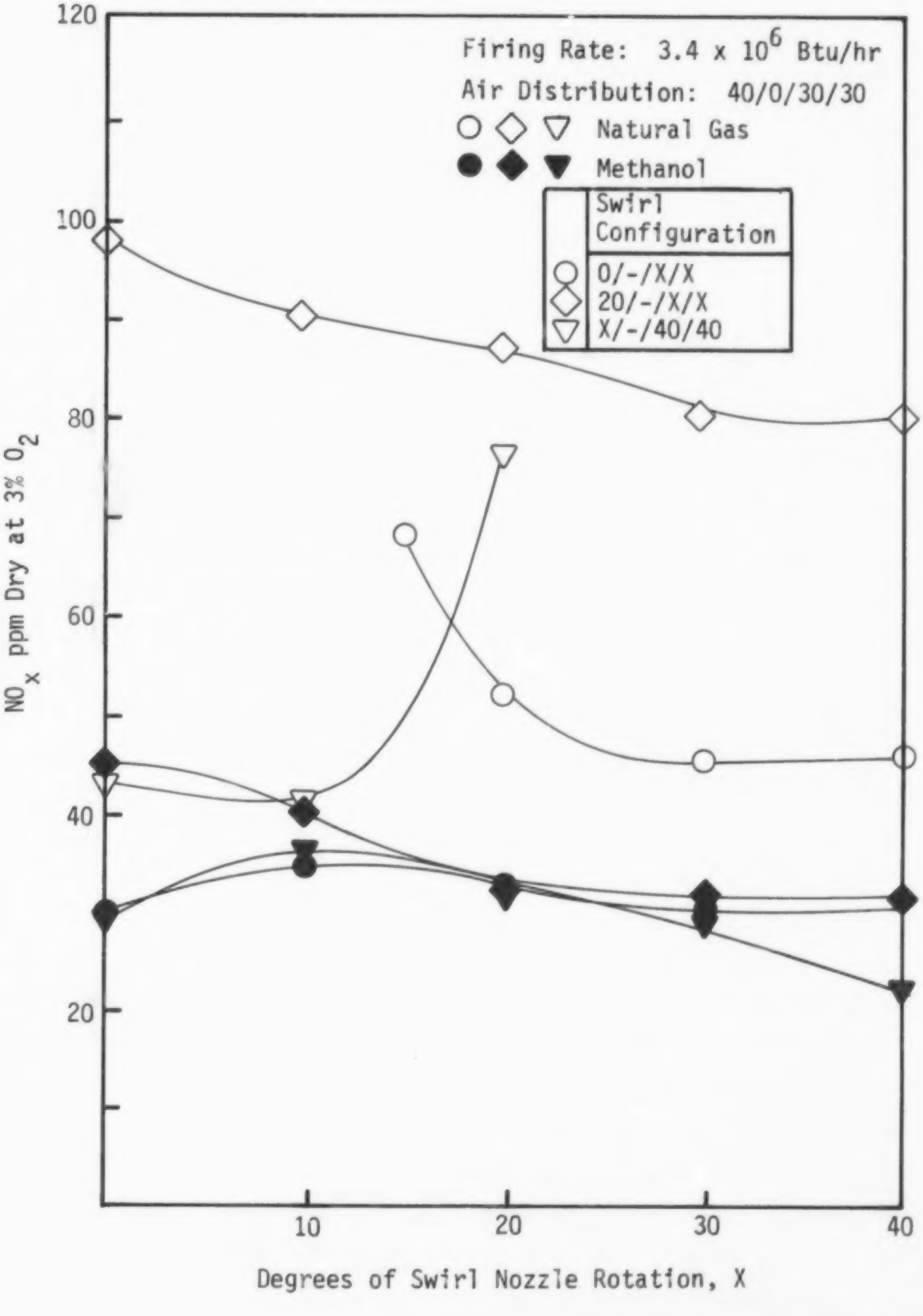


Figure 7-2. The Effect of Swirl Nozzle Rotation on NO_x Emissions with Natural Gas and Methanol for a 40/0/30/30 Air Distribution.

nozzle angle was required on Nos. 3 and 4 to maintain a stable flame with this air distribution, and the NO_x levels measured for both methanol and natural gas were found to be more sensitive to swirl configuration.

A comparison of the range of NO_x emissions obtained under this contract and under Contract 68-02-1500 from a variety of fuels as a function of air/swirl configuration is given in Figure 7-3. The maximum and minimum NO_x measured from a particular fuel at any nozzle position is shown regardless of the air and swirl configuration used to obtain that value. This method of presentation of results then represents the range of NO_x from each fuel over a wide range of tangential/axial momentum. The high nitrogen content No. 6 fuel oil gives the highest NO_x emission and the widest range of measured values. Emission from liquid fuels tend to decrease in descending order of fuel nitrogen content, although for certain burner conditions, emission levels from different fuels are comparable. The range of NO_x emissions observed from natural gas is larger than that shown for propane because a greater number of data points at different test conditions were recorded with propane. Methanol NO_x emissions are the lowest and when displayed on this scale, appear to be independent of air nozzle position.

7.1.2 Commercial Burner

The commercial Ray Burner was used to evaluate NO_x emissions from methanol in relation to other fuels as a function of standard burner operating parameters (excess air level, firing rate, atomization pressure, etc.). The influence of excess air levels on NO_x emissions from two different No. 6 fuel oils, No. 2 fuel oil and methanol is shown in Figure 7-4. NO_x levels (corrected to 3 percent O_2) are displayed as a function of excess O_2 concentrations. The high nitrogen content No. 6 fuel oil exhibits little change in emissions with excess air level. The residual fuel oil with a nitrogen content of 0.2 shows a definite increase in NO_x emissions of almost 75 ppm with increasing excess air level. Limited data for the No. 2 fuel oil shows NO_x emissions at a magnitude of approximately two-thirds the level of those from the low nitrogen content No. 6 oil. Methanol NO_x emissions appear to be independent of excess air level and at a magnitude of less than half that from No. 2 oil.

The effect of firing rate on NO_x from three fuels oils and methanol is shown in Figure 7-5. The difference in NO_x levels between the two residual oils is greatest at high firing rate; the magnitudes approach one another at intermediate firing rates. NO_x emissions from both No. 2 oil and methanol are

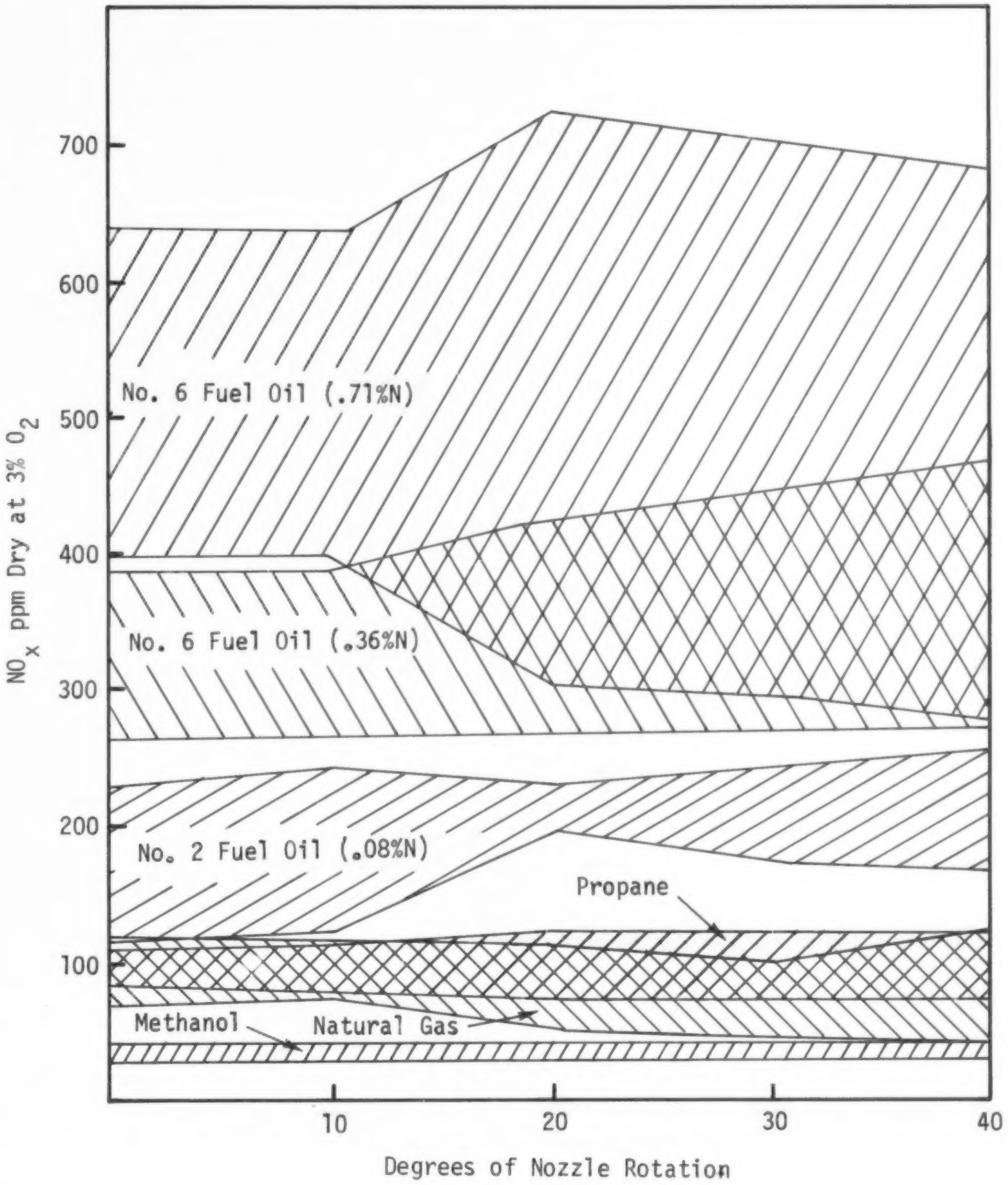


Figure 7-3. Range of NO_x Emissions.

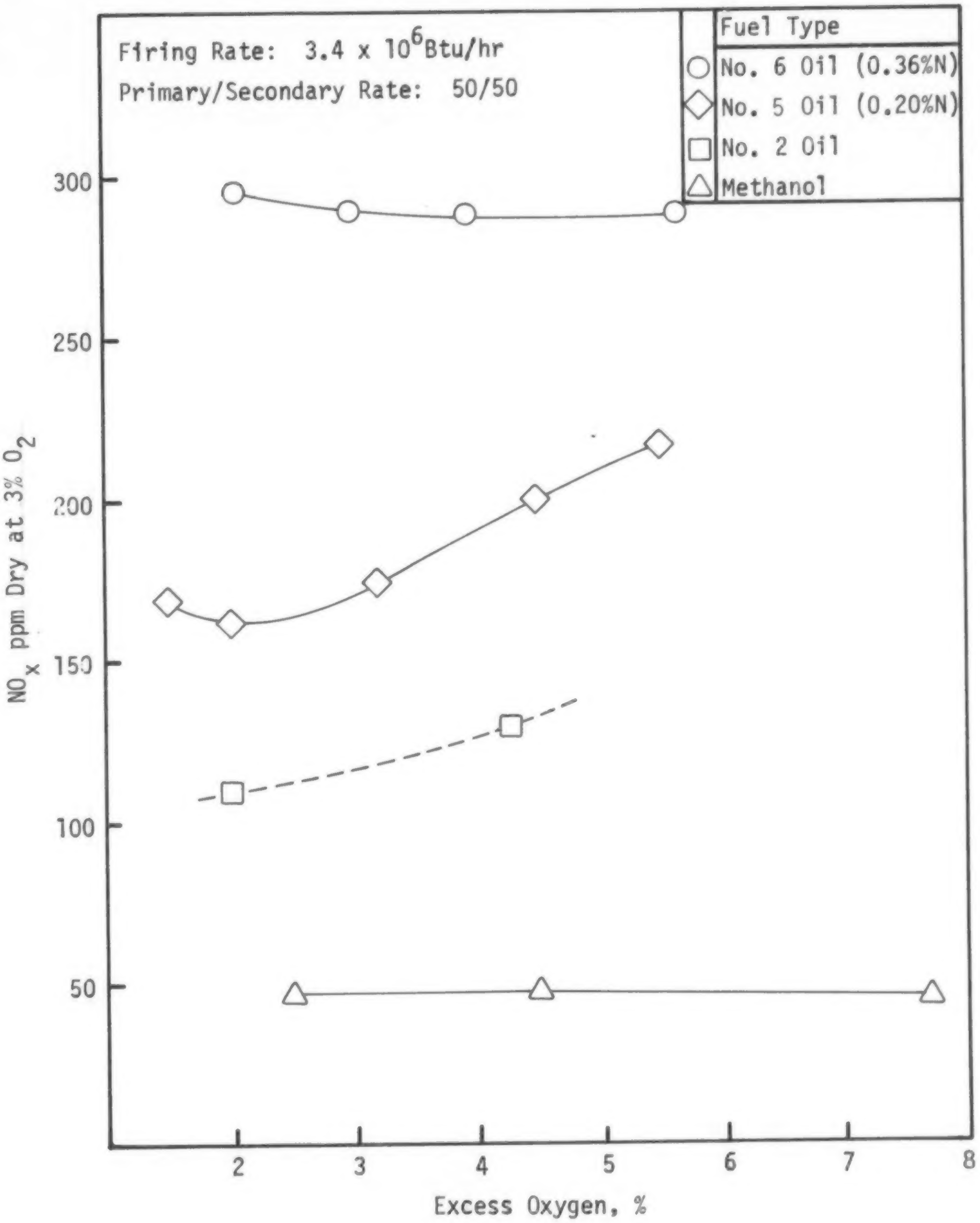


Figure 7-4. The Effect of Excess Air on NO_x Emissions from Various Fuels.

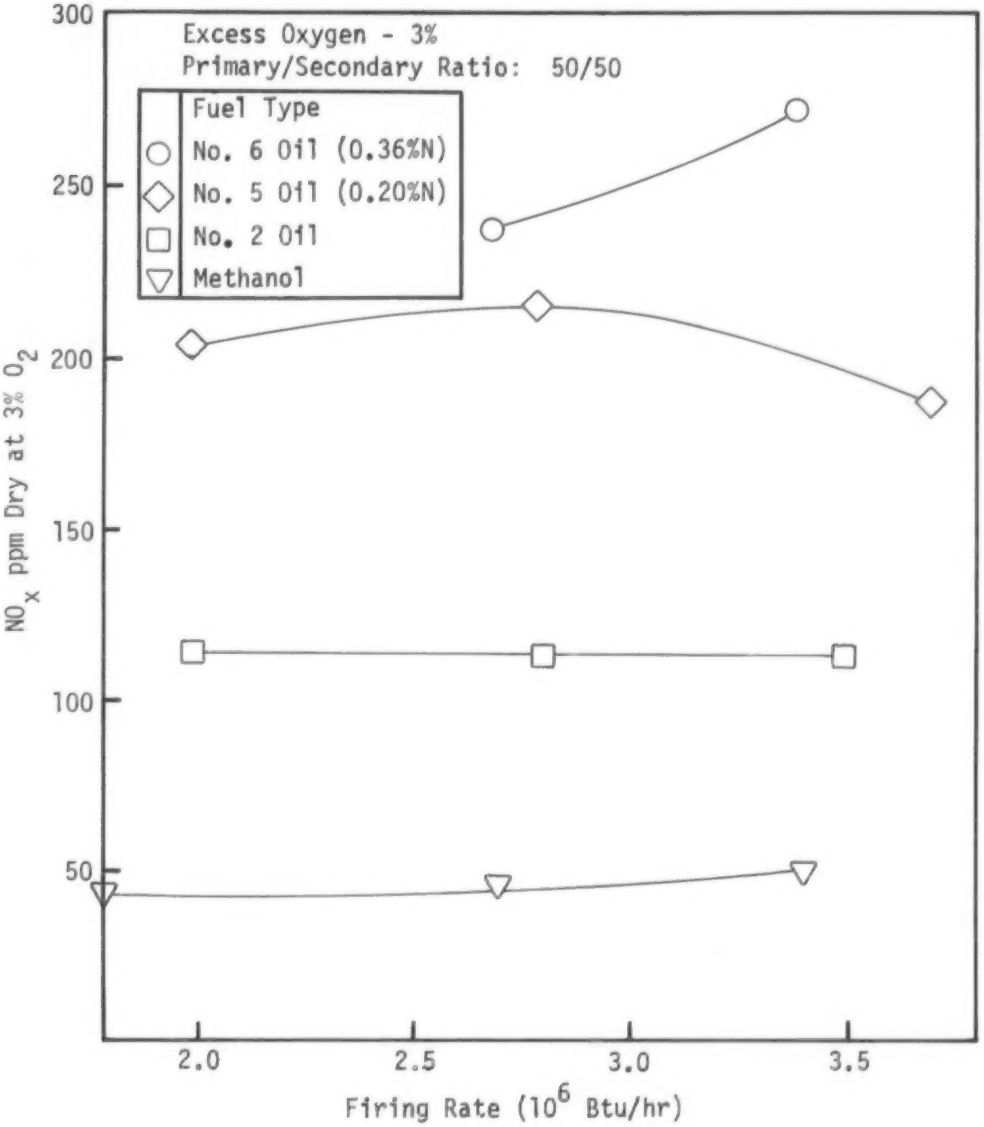


Figure 7-5. The Effect of Firing Rate on NO_x Emissions.

insensitive to firing rates for this particlar burner.

The influence of air atomization pressure on NO_x levels from the various fuels oils and methanol is given in Figure 7-6. NO_x emissions from residual fuel oils appear to be more dependent on air atomization pressure than those from No. 1 fuel oil and methanol. Increasing the atomizing pressure causes a slight decline in NO_x levels for residual fuel oils.

The effect of Primary/Secondary ratio on NO_x emissions from a number of fuels is shown in Figure 7-7. This data was obtained at a fixed firing rate and 17 percent excess air. In addition to the effect of P/S ratio, Figure 7-7 also allows the effect of combustor chamber wall temperature on NO_x emissions to be noted. Data is displayed for No. 6 fuel oil and for methanol with the refractory wall inserts. It should be noted that the presence of the wall liners not only changes the wall temperature, but the inside diameter of the firetube.

7.1.3 Species Concentration Distribution Measured in the Experimental Combustor

The experimental results presented earlier compared flue gas NO_x concentrations resulting from the combustion of different fuels under various operating conditions. Species concentration profiles were measured within the combustion chamber to try to determine whether there were significant differences in the NO_x formation processes between methanol and a conventional fuel. These measurements were made with the probing apparatus described in detail in Reference 17. The concentration of NO, NO_x, CO, CO_2 and O_2 were measured as a function of radial distances at several axial stations. Results are presented for natural gas and methanol for two swirl configurations in Figures 7-8 through 7-11 and the concentrations shown in these figures represent average values for that radius. During these investigations, problems were encountered with the chemiluminescent detector when used in the NO_x mode in the fuel-rich regions of methanol flames. The problem of interference of high concentrations of combustion intermediates (CO, H_2) with the NO_2 converter had been observed on other occasions when the sample contained large concentrations of hydrocarbon gas, carbon monoxide and nitrogen. The anomalies observed in these tests involved the loss of NO/NO_2 from samples taken from the fuel-rich regions of methanol flames when a low temperature (300^oC) carbon (NO_x to NO) converter was used. The same effect was not apparent with samples obtained from the fuel-rich regions of comparable natural gas flames.

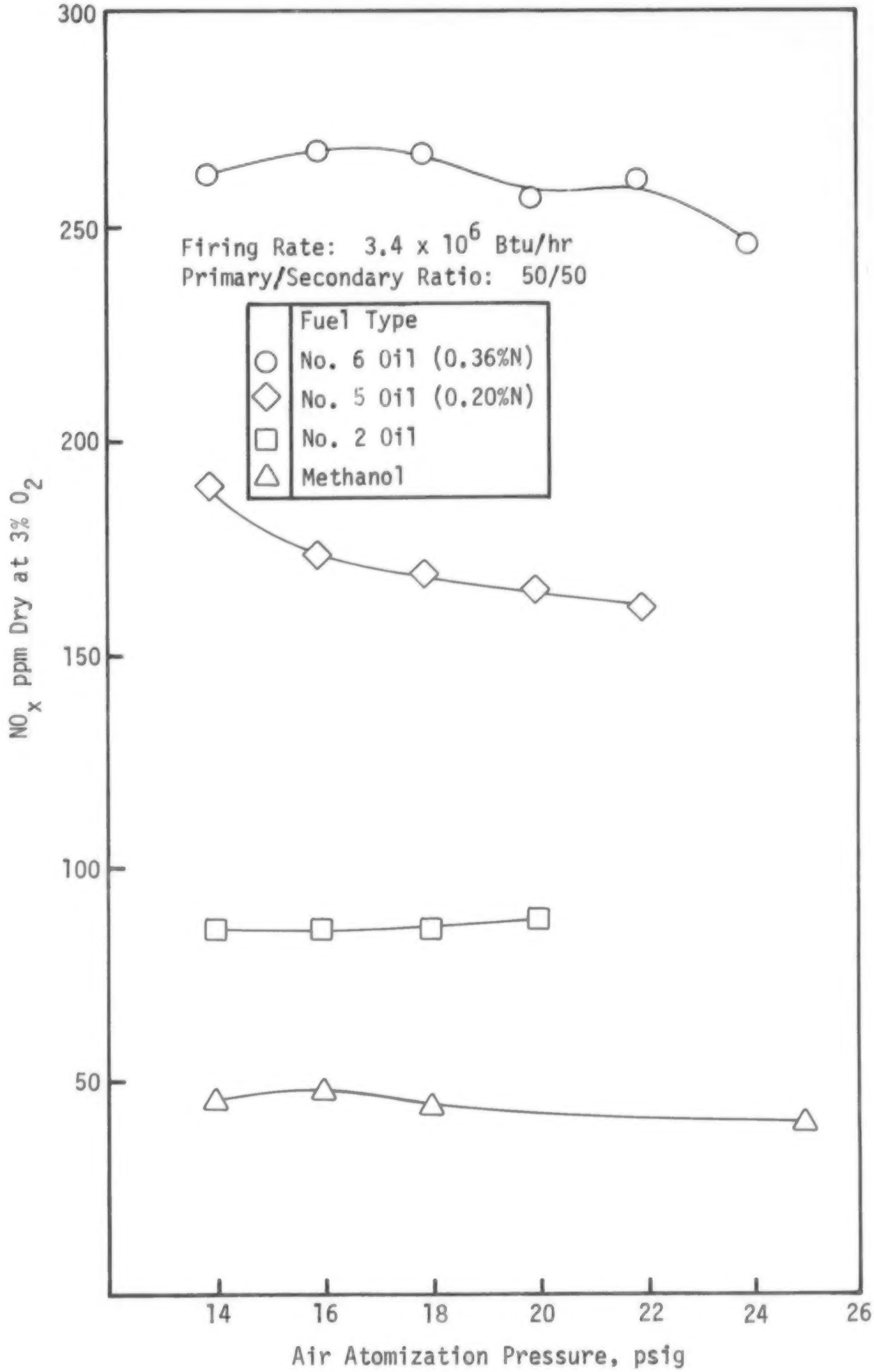


Figure 7-6. The Effect of Air Atomization Pressure on NO$_x$ Emissions.

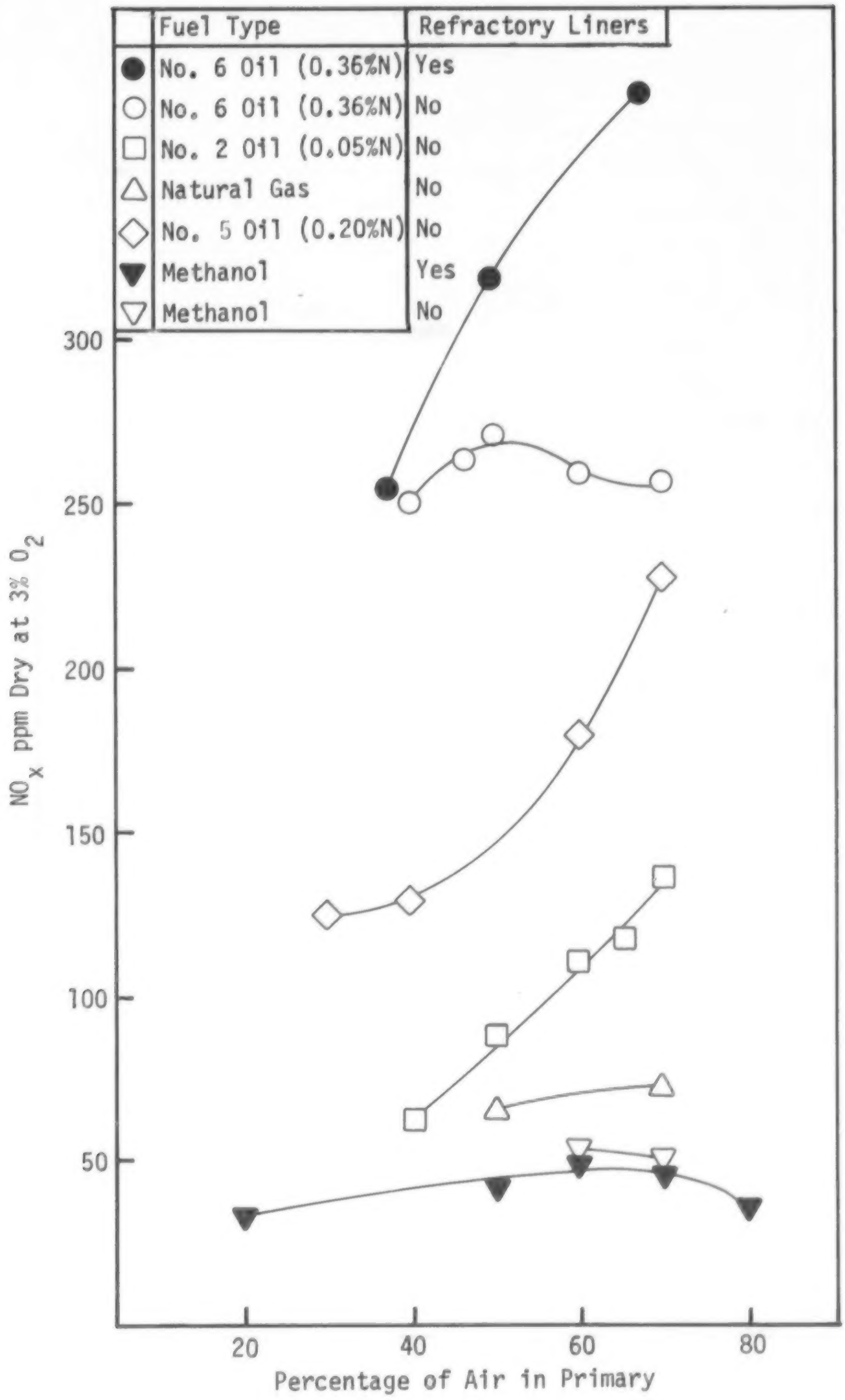


Figure 7-7. The Effect of Primary/Secondary Distribution on NO_x Emissions.

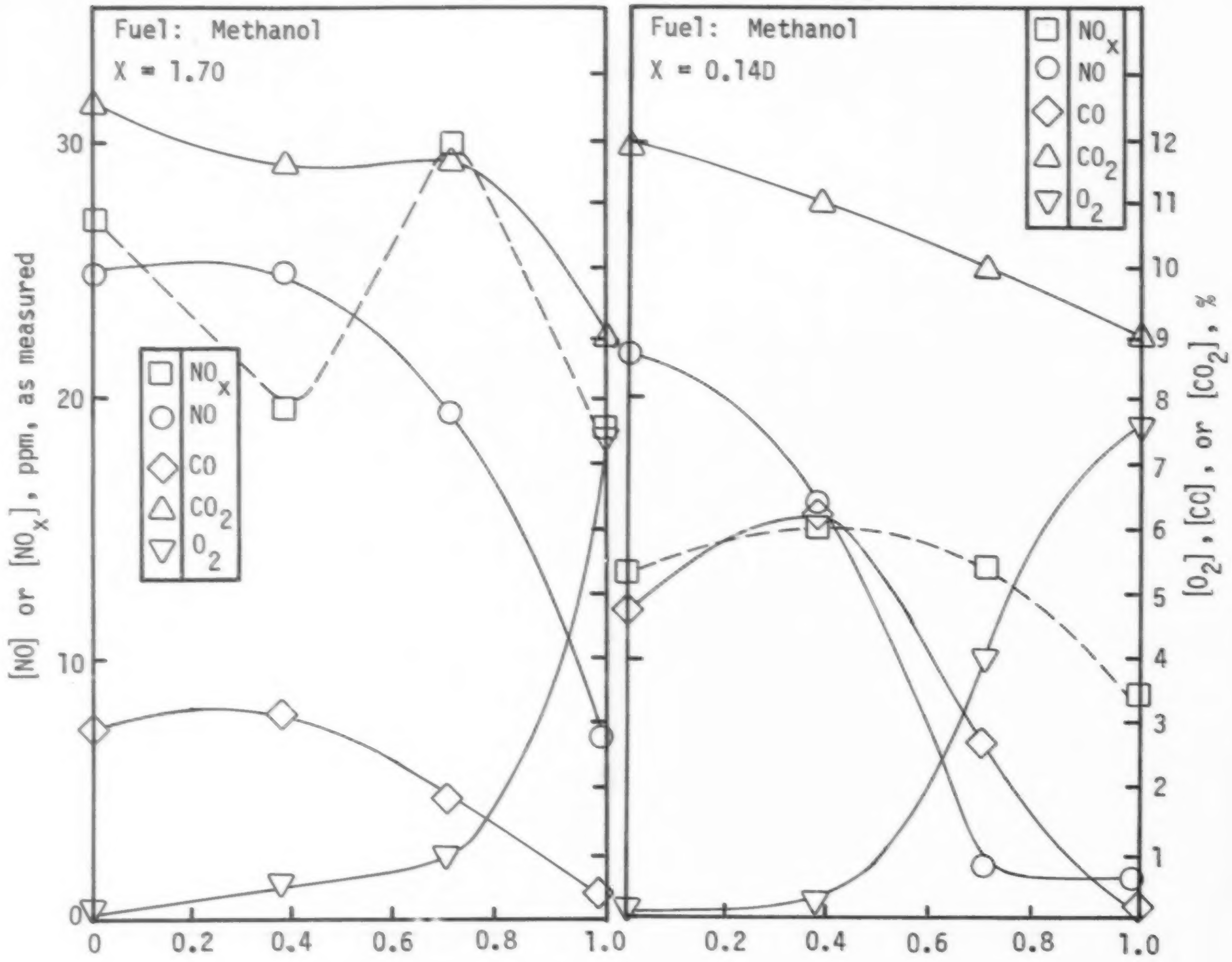


Figure 7-8. Radial Specie Profiles in Firetube at Axial Stations of 0.14D and 1.7D; Methanol Fuel, 31/32 ppm NO/NO$_X$ in Stack, 3% O$_2$ Air/Swirl Configuration: 40/0/30/30, 20/-/40/40.

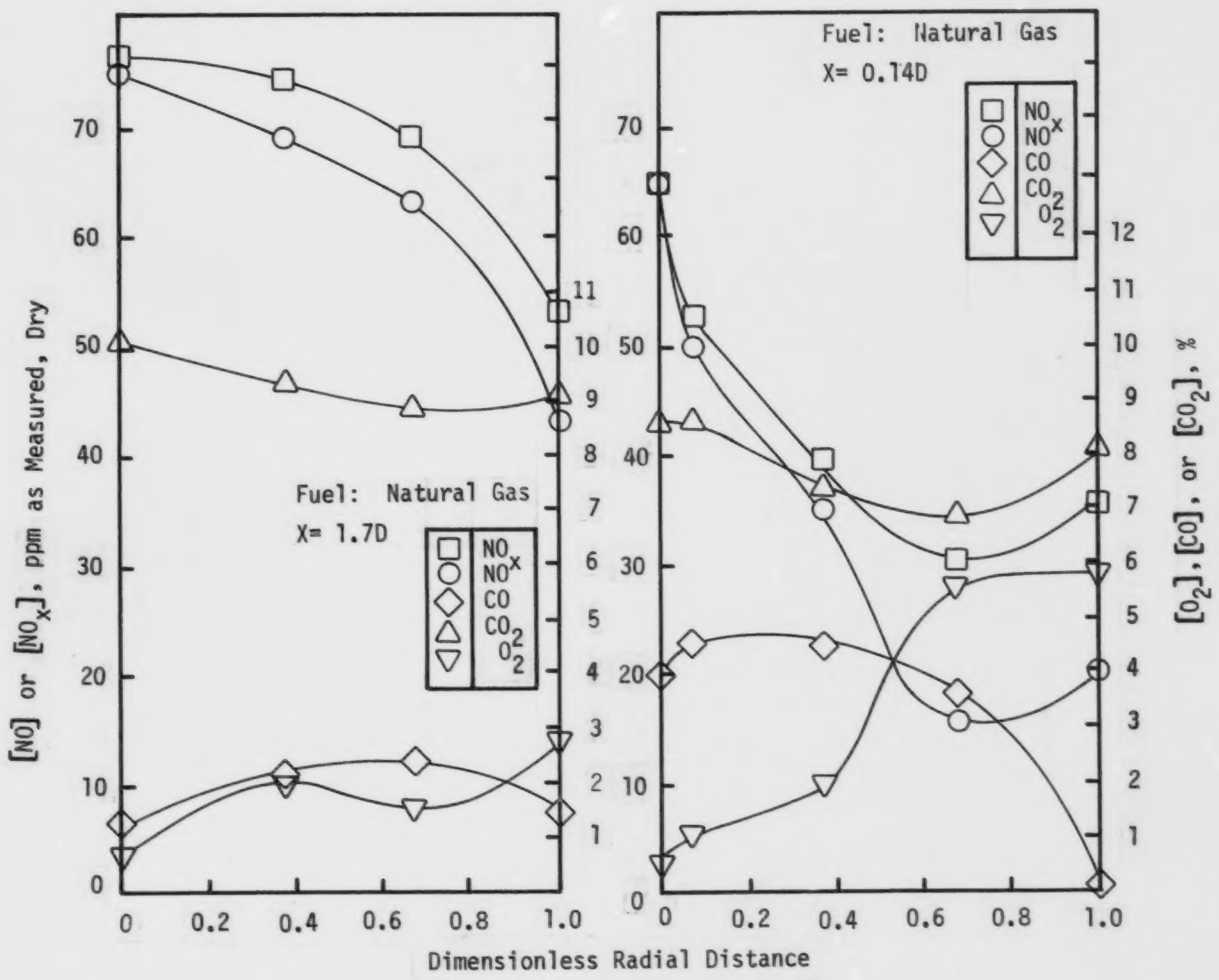


Figure 7-9. Radial Specie Profile in Firetube at Axial Stations of 0.14D and 1.7D; Natural Gas 69/70 ppm NO/NO$_x$ in Stack, 3% O$_2$, Air/Swirl Configuration: 40/0/30/30, 20/-/40/40.

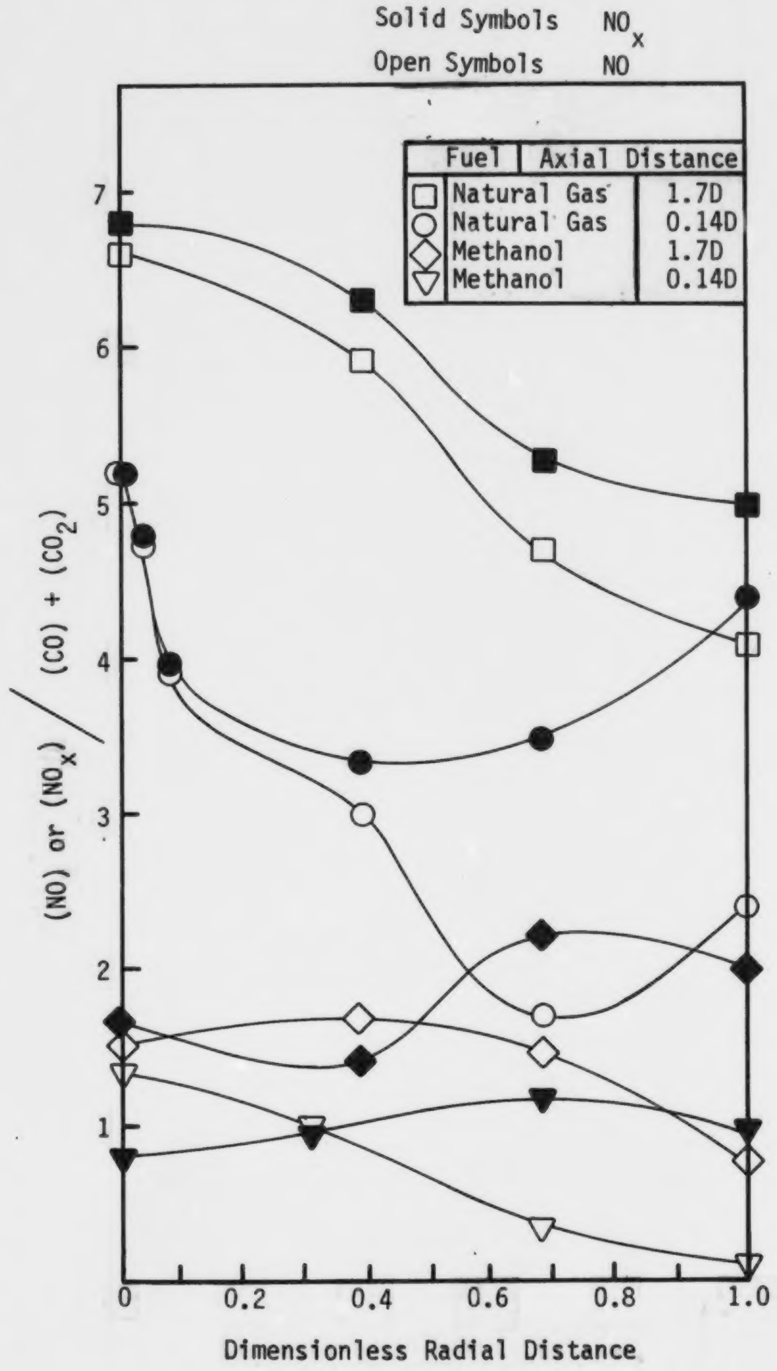


Figure 7-10. Radial NO/NO$_x$ Profiles Normalized to Total CO and CO$_2$ for Methanol and Natural Gas Firing; Axial Stations 0.14D and 1.7D, Air/Swirl Distribution: 40/0/30/30, 20/-/40/40.

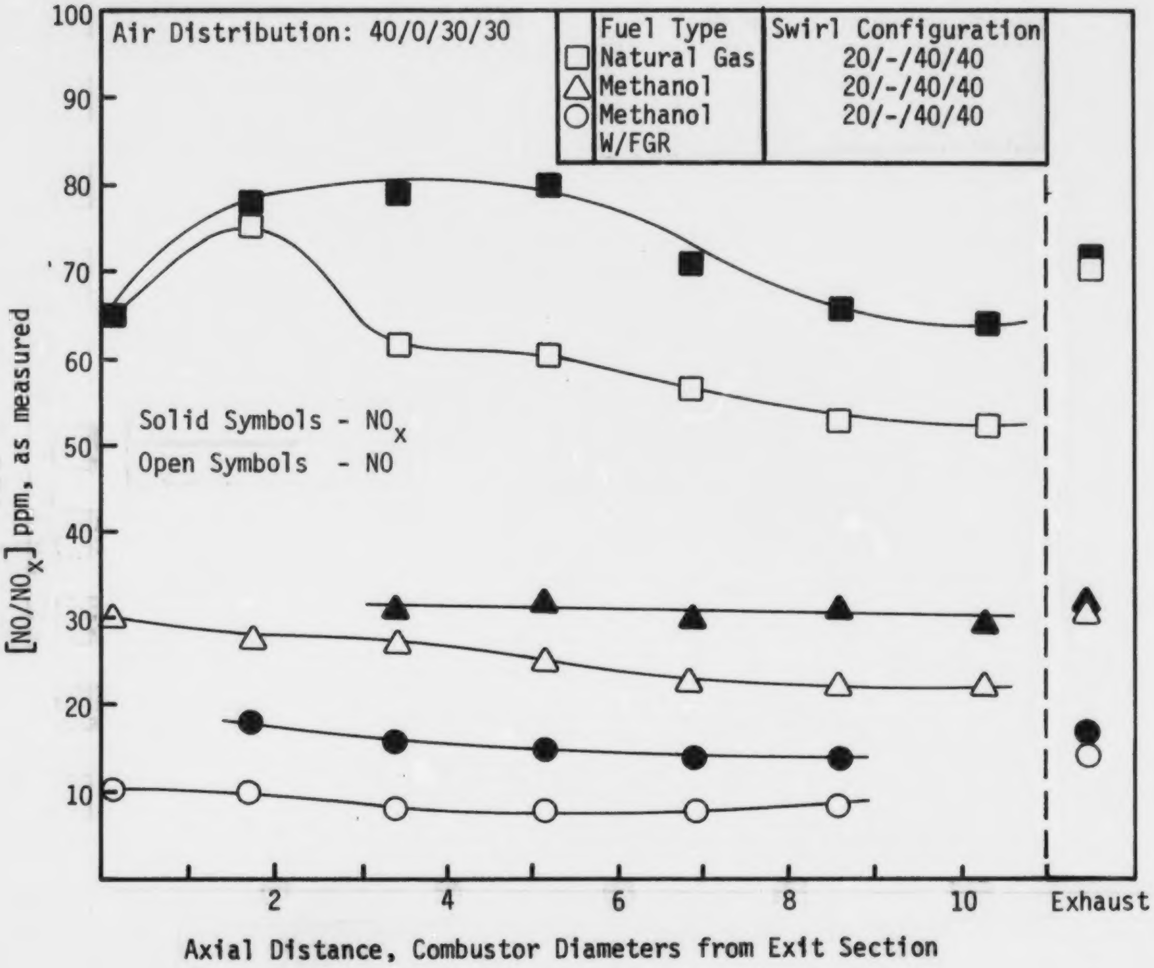


Figure 7-11. Axial NO/NO$_x$ Profiles for Firing with Natural Gas, Methanol, and Methanol with Flue Gas Recirculation; Flue Gas NO/NO$_x$ Stack Conditions: Natural Gas - 73/74, Methanol - 31/32, Methanol with FGR - 18/20.

Average radial profiles for NO, NO_x, CO, CO_2 are given for methanol in Figure 7-8 and natural gas in Figure 6-9. The air/swirl configuration for both flames was 40/0/30/30, 20/-/40/40 and the average radial profiles were obtained at axial stations of 0.41D and 1.7D (D is the combustor diameter) from the fuel injector.

NO profiles for methanol at both axial stations are maximum at the centerline and decay rapidly close to the combustor wall. The peak and average magnitudes are higher at the 1.7D station, indicating continuing NO_x formation after the 0.14D station. For both axial stations, substantial amounts of NO_2 were recorded near the combustor wall. CO levels are maximum off-center at a point where the maximum fuel concentration would be expected, and decrease at the wall. The magnitude of CO decreases from the 0.14D to 1.7D station. Oxygen profiles for both stations are very low (0.3 percent) in the vicinity of the centerline and increase sharply as the wall is approached. CO_2 profiles at both stations are maximum at the centerline and decrease as the wall is approached.

Average species profiles are illustrated for natural gas in Figure 7-9. At 0.14D, NO/NO_x peaks at the centerline and drops abruptly at off-center positions. Large quantities of NO_2 are measured close to the wall, similar to the levels observed with methanol. At X=1.7D, the profile is more uniform and the average value is much higher than 0.14D, indicating continued NO/NO_x formation. CO is maximum off-center and decreases as the wall is approahced for both axial stations. CO_2 profiles are relatively uniform and increase slightly at the 1.7D station. Oxygen is measured in very low concentrations at the centerline and increases abruptly at the wall. As with methanol, the profile is flatter at the station further downstream. At both axial locations, the measured O_2 concentrations are much lower with methanol than with natural gas (<<0.1 percent compared to approximately 0.5 percent). This reflects a change in the mixing pattern of the two fuels due to differences in the method of fuel injection.

The NO/NO_x prifiles from methanol and natural gas can be best compared by considering the amount of NO/NO_x normalized to the total carbon concentrations (i.e., CO + CO_2) which tend to eliminate concentration changes due to dilution. The NO/NO_x profiles for both fuels at both axial stations are displayed in this manner in Figure 7-10. A comparison of NO/NO_x from methanol and natural gas in this fashion shows:

- The decrease in NO/NO_x profiles in the vicinity of the wall is less severe due to elimination of the oxygen diluent effect.
- The NO/NO_x profiles for both natural gas and methanol increase in magnitude from 0.14D to 1.7D indicating continued NO/NO_x formation. The increase is much larger with natural gas than with methanol, suggesting the methanol NO formation zone is smaller.

The NO/NO_x concentration measured at the centerline at several axial positions is illustrated in Figure 7-11 for natural gas and methanol. Total NO_x is displayed only at axial stations where anomolous behavior was not observed. The difference is size of the NO formation zone between natural gas and methanol is apparent. Axial NO_x concentrations are still increasing with natural gas five combustor diameters from the nozzle exit section. Also the concentration of NO_2 relative to total NO_x is higher with methanol than with natural gas at the exit of the combustion chamber.

7.2 Emission Characteristics of Field Boilers as a Function of Fuel Type

The schedule of field testing at the Essex County Correction Center (ECCC) is presented in Appendix A and because the tests were carried out over a relatively long period of time, it is possible to assess the effect of any long-term changes in boiler performance. This comparison is complicated for fuels containing nitrogen since over a long period of time, there can be no guarantee of constant fuel nitrogen content. Comparison of results from heavy fuel oil firing then show not only the effect of any long term changes in system performance, but also changes in fuel composition. NO_x emissions from these tests and other field results are presented corrected to 0% O_2 for a dry sample.

Figure 7-12 presents the influence of excess air on NO_x emissions from the firetube boiler firing natural gas for two trial dates. The results which were obtained at similar loads in the Spring of 1974 (carried out under Contract 68-02-0222) and the Spring of 1976 trials are almost identical, suggesting no gross change in system operation occurred over the time span. NO_x emissions from the firetube boiler firing No. 5 fuel oil are presented as a function of excess air level for the three trials dates in Figure 7-13. At both loads, the Spring 1974 measurements are the highest with Spring 1975 and Spring 1976 levels almost equal. The nitrogen content of the fuel oil during these trials was reported at 0.11, 0.04, and 0.12 wt percent for the Spring 1974, Spring 1975 and Spring 1976 trials respectively. The low nitrogen content of the fuel during Spring

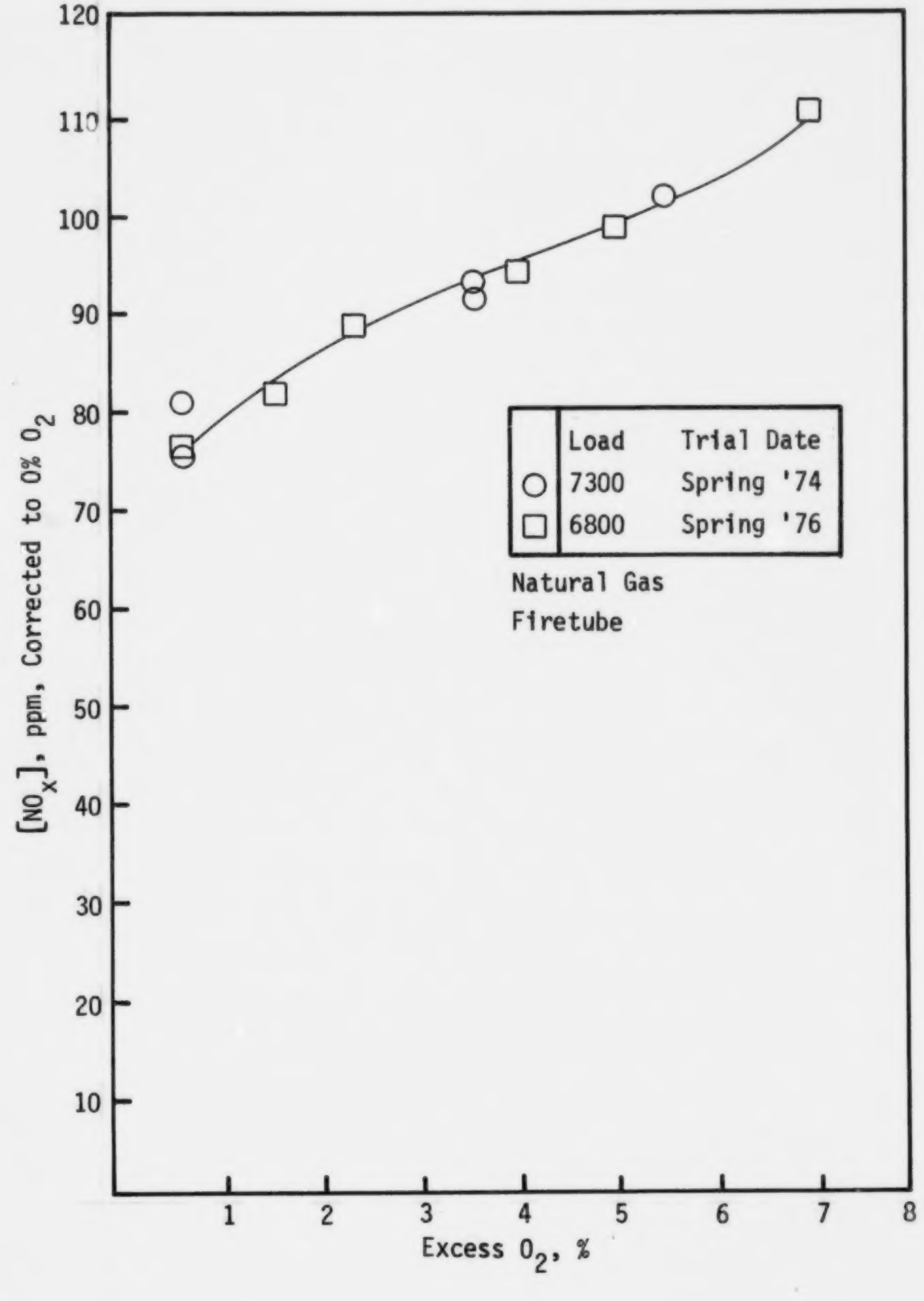


Figure 7-12. Reproducibility of NO_x Emission Levels with Natural Gas in the Firetube Boiler.

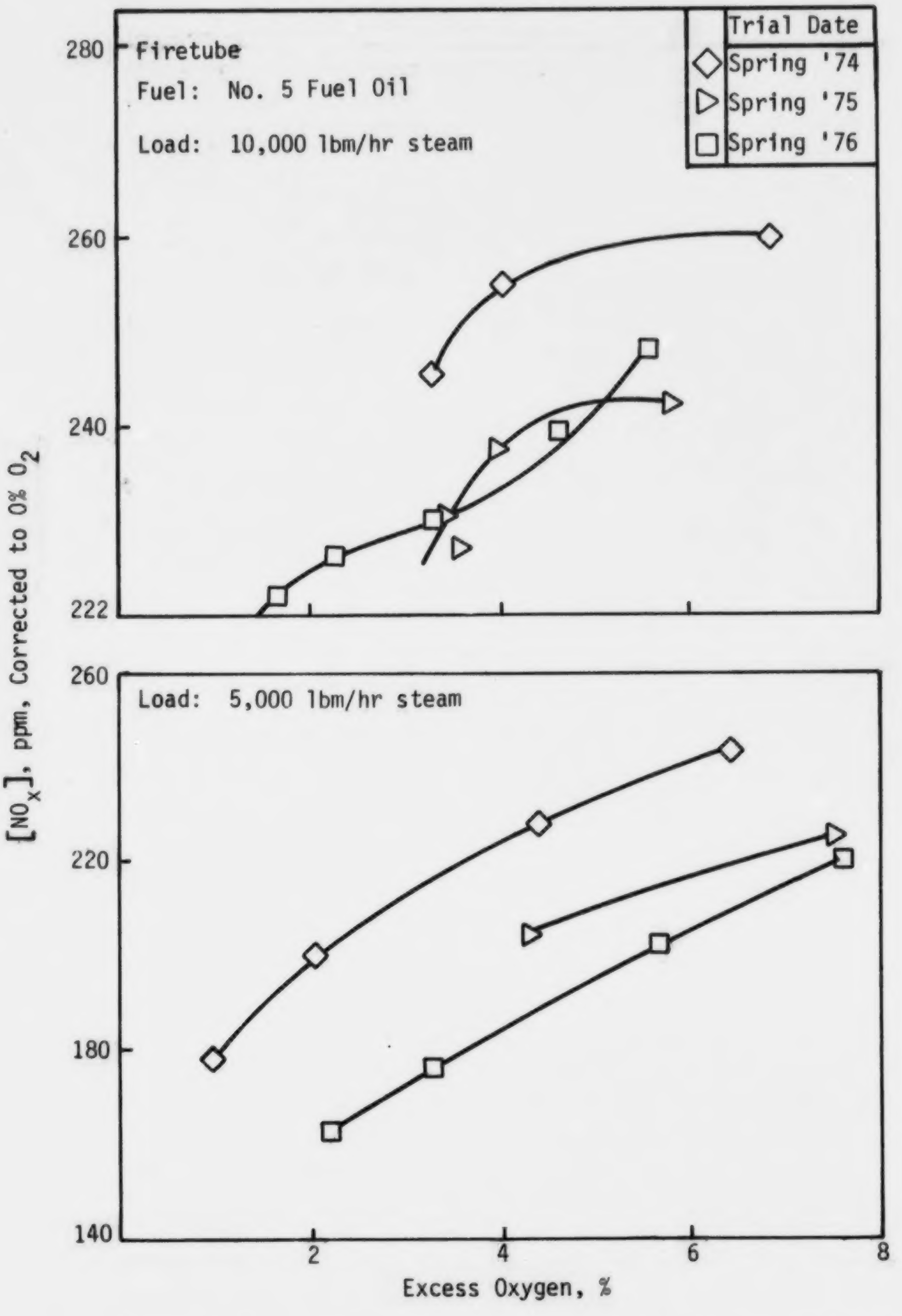


Figure 7-13. Variation in NO_x Emission Levels for Different Trial Dates.

1975 trials is totally inconsistent with the observed NO_x levels and strongly suggest an error in the analysis as there is no apparent reason for a variation in boiler performance.

Data for methanol firing in the firetube boiler is given in Figures 7-14 and 7-15. The only modification necessary was the use of a fuel nozzle of larger flow capacity to compensate for the lower volumetric heating value of methanol. Figure 7-14 presents NO_x levels as a function of excess air for three loads and the original and "oversized" nozzles. At low load (4,000 lbm/hr), NO_x from the oversized nozzle is higher at moderate (3-6 percent) excess air, but drops off rapidly to below that of the standard nozzle at high excess air. The highest overall NO_x concentration was obtained at the highest load (10,000 lbm/hr) with the oversized nozzle. The highest load attainable from the standard nozzle firing methanol was 6,000 lbm/hr.

In Section 4, fuel properties of methanol were compared to those of fuel oil. Those fuel properties which exhibit large differences (i.e., vapor pressure, latent heat, and their dependence on temperature) raise the possibility that fuel temperature may have a substantial impact on the rate of fuel/air mixing and therefore NO_x formation. Martin and Heap(15) have suggested this from visual observations of burning methanol sprays. The effect of fuel temperature on NO_x emissions was investigated in the firetube boiler and the results are presented in Figure 7-15. The fuel temperature was varied between ambient (70^{0}F) and approximately 105^{0}F for these tests. Over this modest temperature range fuel temperature appears to have only a minor influence upon NO_x emissions and the effect is dependent upon nozzle size and load. At medium load, virtually no effect of fuel temperature on NO_x is noted when the original nozzle was used. With the oversize nozzle at medium load, slightly higher NO_x emissions were observed with the fuel at ambient temperature, except at very high excess air levels when there was a significant decrease in emission. At low load with the oversized nozzle, higher NO_x emissions were obtained with the ambient temperature fuel over the entire excess air range.

A comparison of NO_x emissions from the firetube boiler as a function of excess oxygen level for fuel oil, natural gas, and methanol firing is given in Figure 7-16. The effect of excess air level on NO_x emissions from methanol and natural gas is small compared to its effect with fuel oil. The large increase in NO_x with increasing excess air is most probably due to the increasing

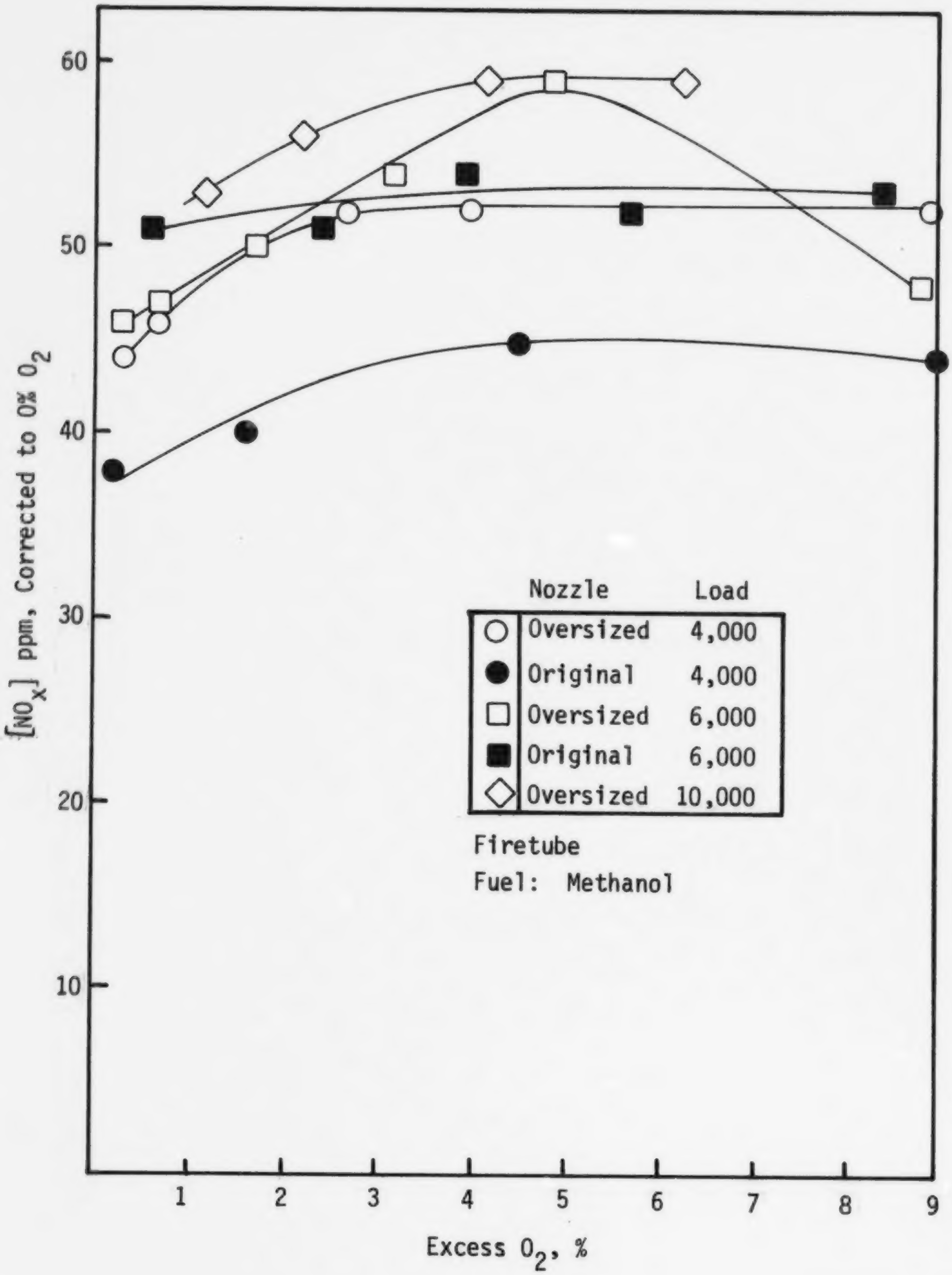


Figure 7-14. The Effect of Excess Air on NO_x Emissions as a Function of Nozzle Size and Loads.

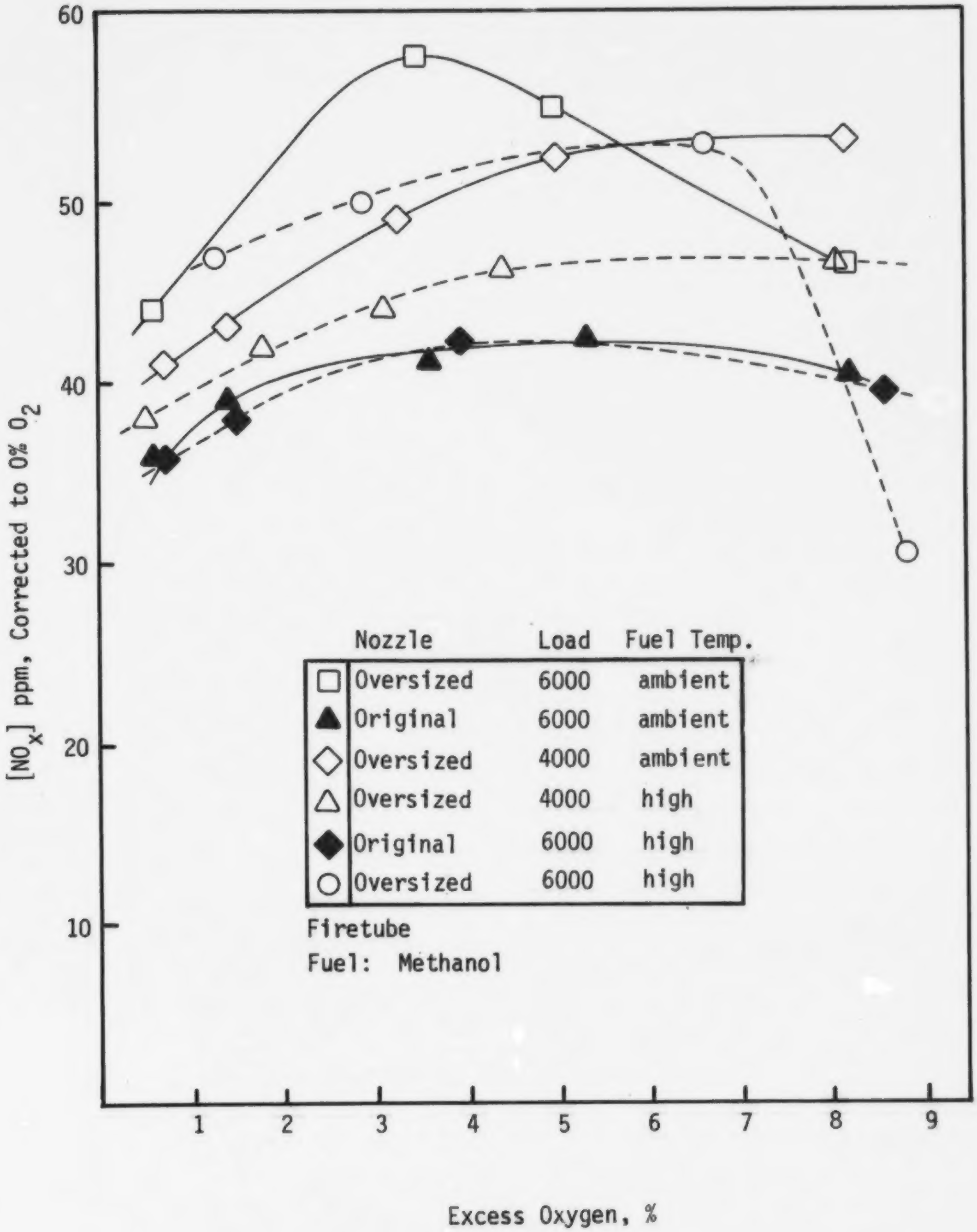


Figure 7-15. The Effect of Excess Oxygen on NO$_x$ Emissions as a Function of Fuel Temperature, Nozzle Size, and Load.

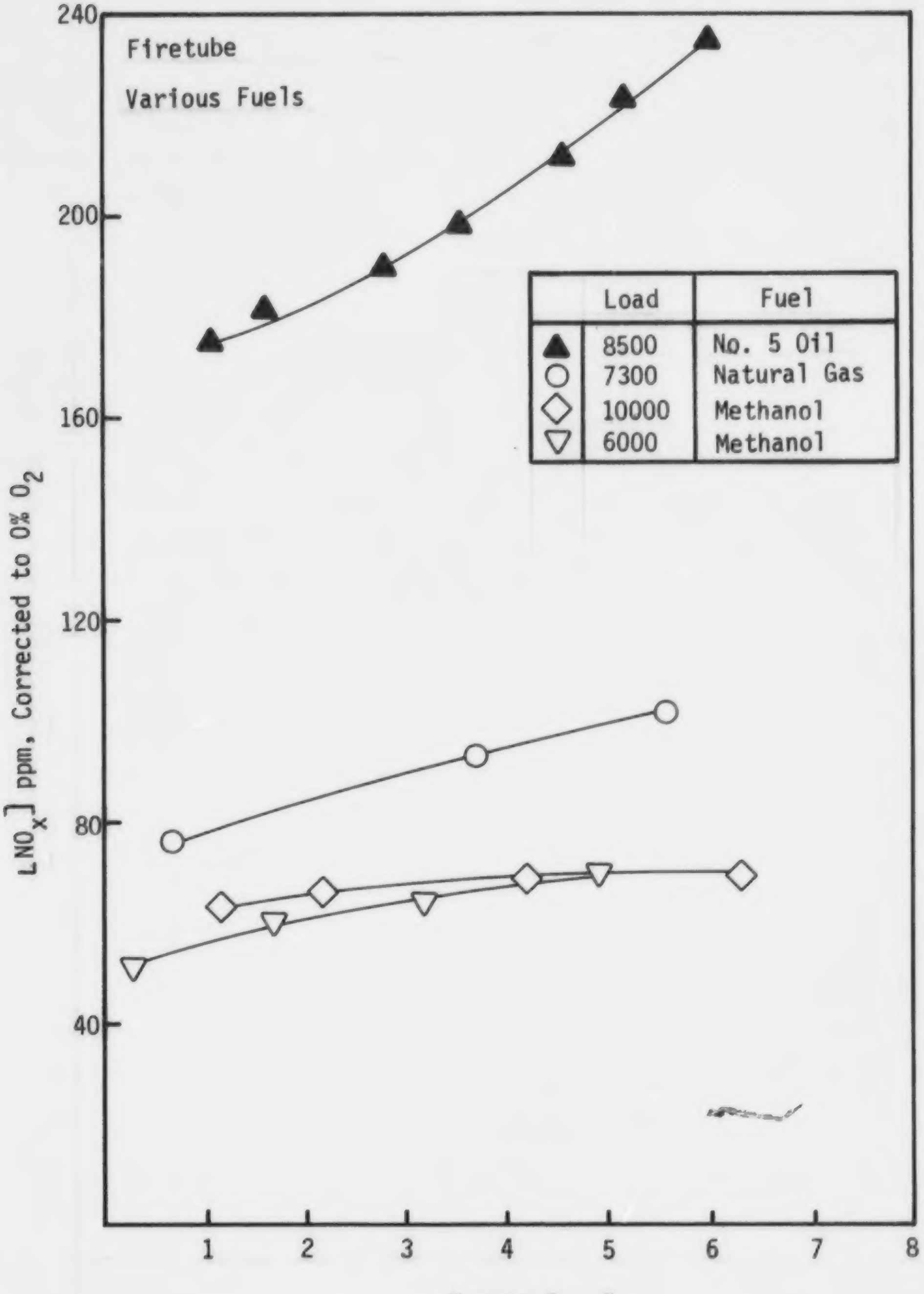


Figure 7-16. The Effect of Excess Oxygen on NO$_x$ Emissions from the Firetube as a Function of Fuel Type and Load.

contribution of fuel-bound nitrogen to total NO_x level with the increased availability of oxygen.

Repeat NO_x emission levels are available for both the watertube and firetube boilers covering a wide time span, but there is no recognizable pattern to the observations. Data for NO_x levels from the watertube are presented in Figures 7-17 through 7-20. The effect of excess air level on NO_x emission from natural gas firing for the three trial dates are given in Figure 7-17. The Spring 1975 and Spring 1976 levels show reasonable agreement, however, emissions measured during the Spring 1974 trials (carried out under Contract 68-02-0222) are lower than those measured later except at high excess air levels. Modifications to the watertube boiler for installation of FGR hardware were made between thse two trial dates; however, it is unlikely that these modifications were responsible for the observed differences. NO_x emissions with oil firing for different trial dates are given in Figure 7-18. The emissions are similar at low excess air levels but differ significantly as higher levels are approached. The nitrogen content of the fuel oil for the 1974, 1975 and 1976 trials was 0.11, 0.04, and 0.12 wt. percent respectively although there is some question concerning the accuracy of the fuel analysis.

A comparison of NO_x levels from the watertube as a function of excess air level for methanol, natural gas, and fuel oil is given in Figure 7-19. The relative differences in NO_x emissions are similar to those observed in the firetube boiler. The influence of excess air level on NO_x emission from the watertube boiler is similar to that observed with the firetube boiler. NO_x emissions with fuel oil firing exhibit a stronger dependence in excess air level than either with either natural gas or methanol.

The atomizing medium and atomizing fluid pressure could be readily varied with the watertube boiler and the influence of these parameters on NO_x emission at both low and high excess air levels is given in Figure 7-20. For both excess air levels, lower NO_x emissions were generally obtained with air atomization.

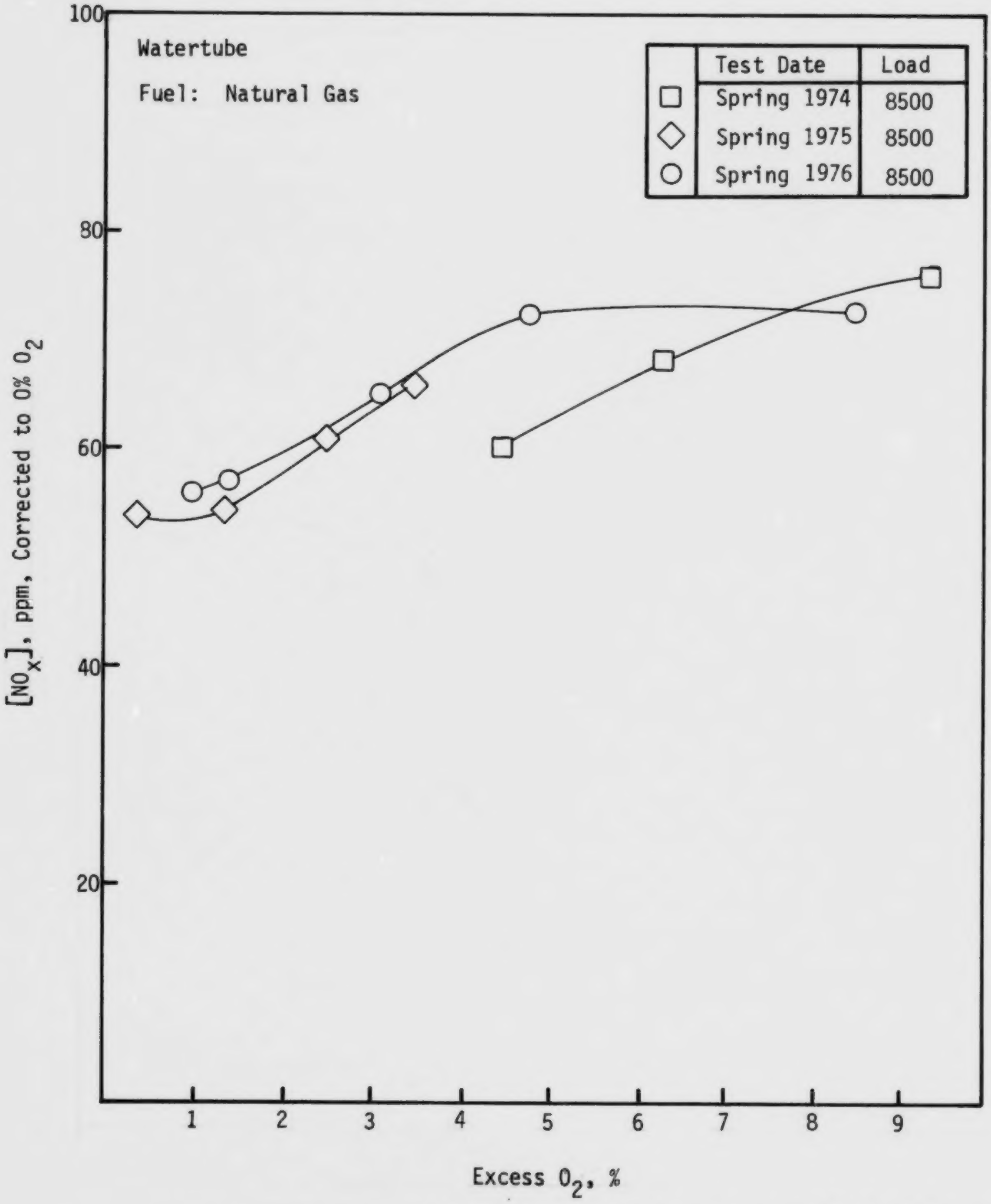


Figure 7-17. The Effect of Excess Oxygen Concentration on NO_x Emissions for Three Trial Dates.

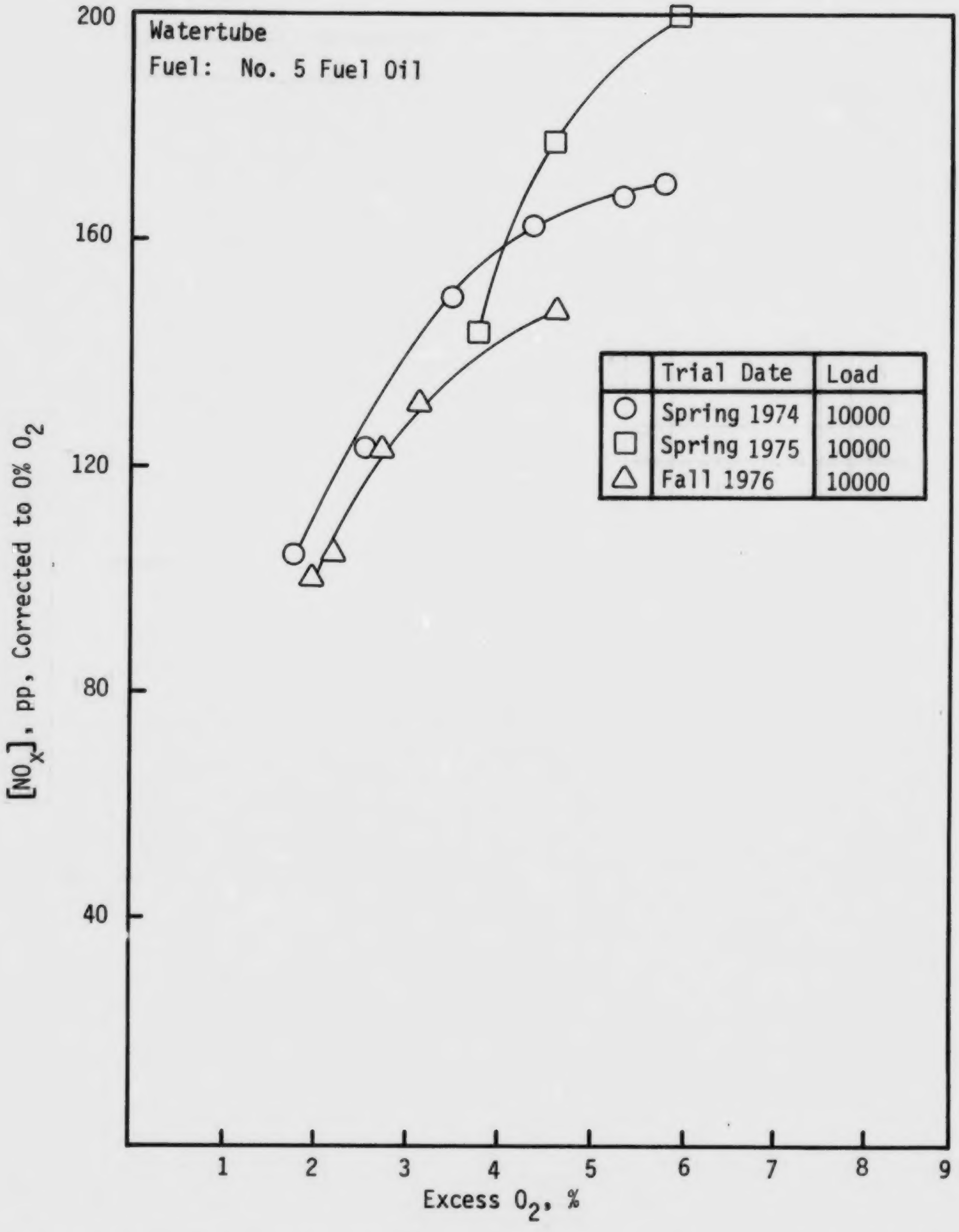


Figure 7-18. The Effect of Excess Oxygen on NO_x Emissions from the Watertube Boiler for No. 5 Fuel Oil Firing for Various Trial Dates.

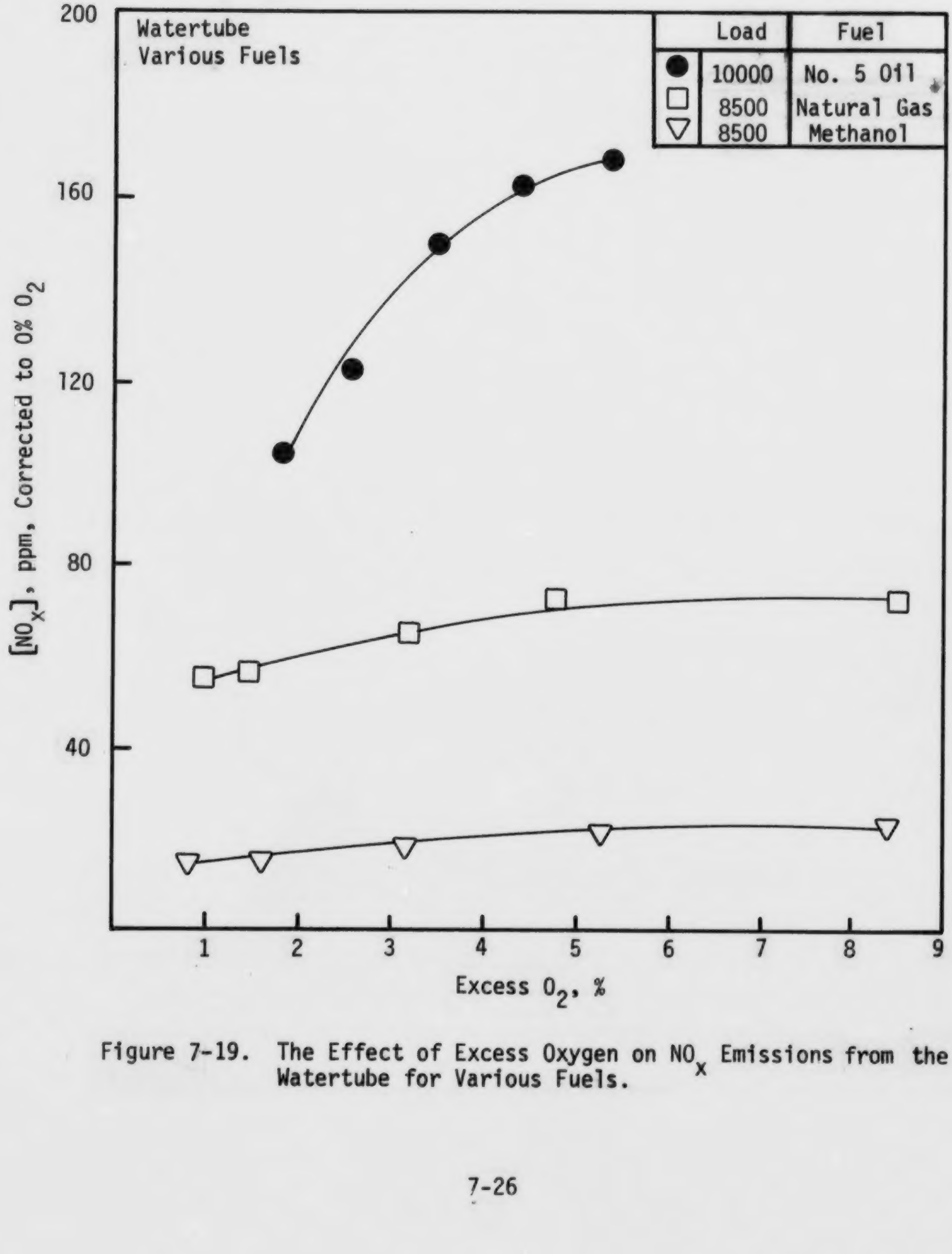


Figure 7-19. The Effect of Excess Oxygen on NO_x Emissions from the Watertube for Various Fuels.

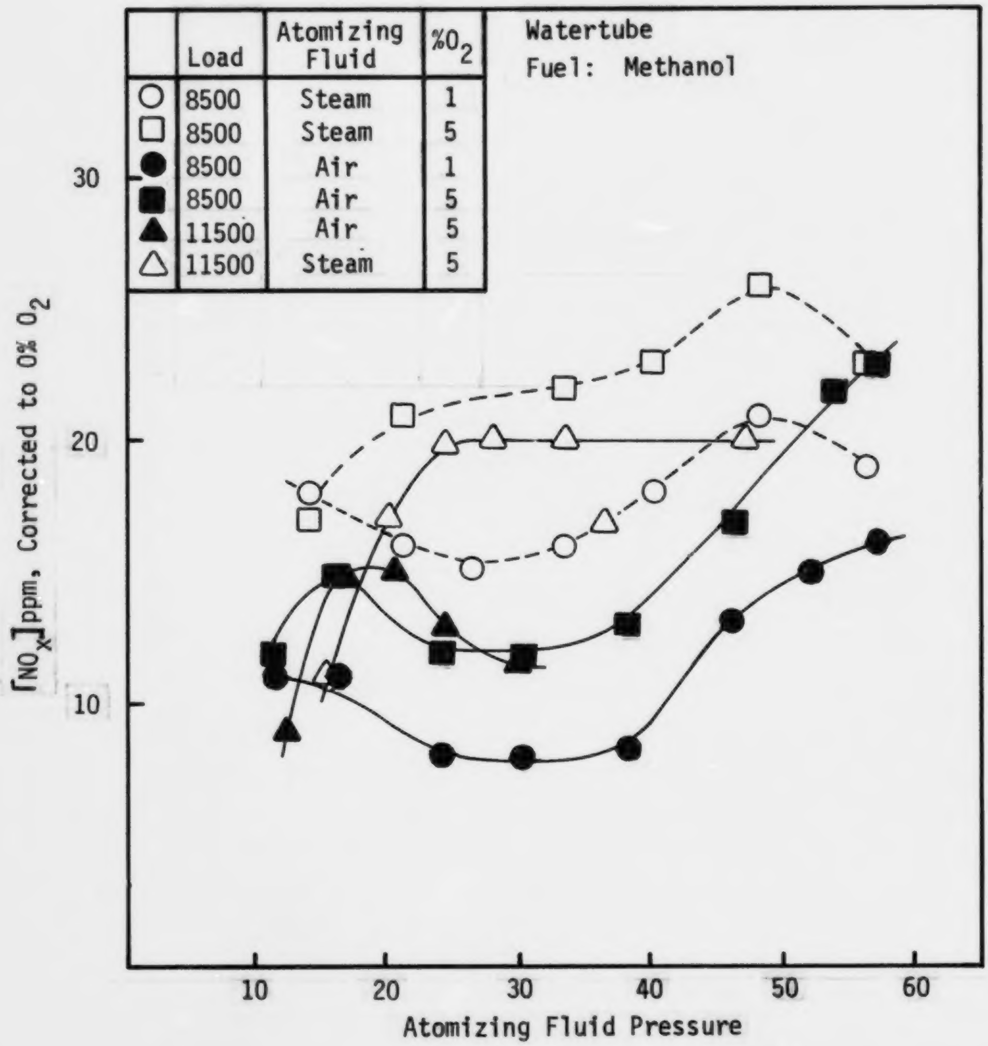


Figure 7-20. The Effect of Atomizing Fluid Pressure on NO_x Emissions from the Watertube for Methanol Firing for Steam and Air Atomization and Low and High Excess Oxygen.

8.0 POLLUTANT CONTROL TECHNIQUES AND BOILER PERFORMANCE

Data concerning the effect of combustion process modifications and fuel variations on pollutant emissins from package boilers has been presented and discussed. These methods of pollutant emission control must be compared on a common basis in terms of cost-effectiveness. This requires that the potential reduction in emissions offered by each technique be balanced against the total costs associated with its use.

The potential costs of implementation of each control technique would be due to the following:

- The capital costs of design, fabrication, and retrofitting the necessary system hardware;
- The cost of additional power requirements and maintenance to operate the hardware;
- Increased fuel costs due to a decrease in thermal efficiency; and
- Potential structural problems cause of variations in the heat absorption rate at the walls of the combustion chamber and possible increases in fireside corrosion rates with liquid fuels.

The running costs associated with steam generation are generally more significant than initial hardware costs and therefore, the most serious cost impact associated with pollutant control would be associated with a decrease in thermal efficiency.

8.1 Thermal Efficiency

The thermal efficiency of steam generation is defined as:

$$\eta = \frac{\text{available energy in steam}}{\text{input energy}} \times 100$$

If the losses from the steam generating system are known, the thermal efficiency may be expressed as

$$\eta = 100 - \frac{\text{energy lost from system}}{\text{input energy}}$$

The ASME Heat Loss Method(18) was used to calculate the thermal efficiency for the boilers under consideration.

The influence of fuel type on thermal efficiency is predominantly due to the effect of combustion product gas composition and the loss of enthalpy associated with the discharged combustion products. The energy in the flue gas is in the form of the latent heat of vaporization of the gas phase water and the heat capacity of the high temperature products. The major effect of fuel type on heat loss is in the amount of water produced per unit energy released and the subsequent latent heat losses since exit temperatures are above the dew point. This is dependent on both the carbon/hydrogen (C/H) ratio of the fuel and the fuel heat content per unit mass of combustion products. A fuel with a higher hydrogen content will produce more water in the stack gas per mole of carbon and thus have higher latent heat losses. If the heating value of the fuel per mass of combustion products is low, the higher fuel and air throughput rates required for a given heat release will mean greater discharge rates of effluent gases and therefore increased stack losses.

The calculated maximum thermal efficiency of a hypothetical boiler based solely on latent and convective losses is shown in Figures 8-1a and 8-1b for three fuels. Figure 8-1a gives the calculated thermal efficiency as a function of excess air levels for a constant flue gas temperature difference of $300^{\circ}F$ from ambient conditions and Figure 8-1b gives the thermal efficiency as a function of flue gas temperature difference for a zero excess air level. The difference in calculated thermal efficiency between fuel oil and natural gas is mostly attributable to the differences in C/H ratio (approximately 0.6 for No. 6 oil and 0.25 for methane) changing the latent heat losses. A significant difference is noted between natural gas and methanol, although the C/H ratios are identical (0.25). This is because the calorific value of methanol is much less than that of methane and a higher throughput of fuel is required. The mass of combustion products per unit time leaving the stack is greater, as are the latent and convective heat losses.

The baseline thermal efficiency measured for both the firetube and watertube boiler firing fuel oil is presented in Figure 8-2 as a function of excess oxygen concentration. At equivalent firing rates, the thermal efficiency of both boilers is similar in both magnitude and response to excess oxygen concentration. The magnitude of thermal efficiency is less than the theoretical values shown in Figure 8-2 due to the inclusion of radiant and unaccounted losses in the heat loss calculation method. The effect of load on thermal efficiency for the firetube can be determined from Figure 8-2; a decrease in firing rate of approximately 40 percent is responsible for an increase in thermal efficiency by an average of one percentage point.

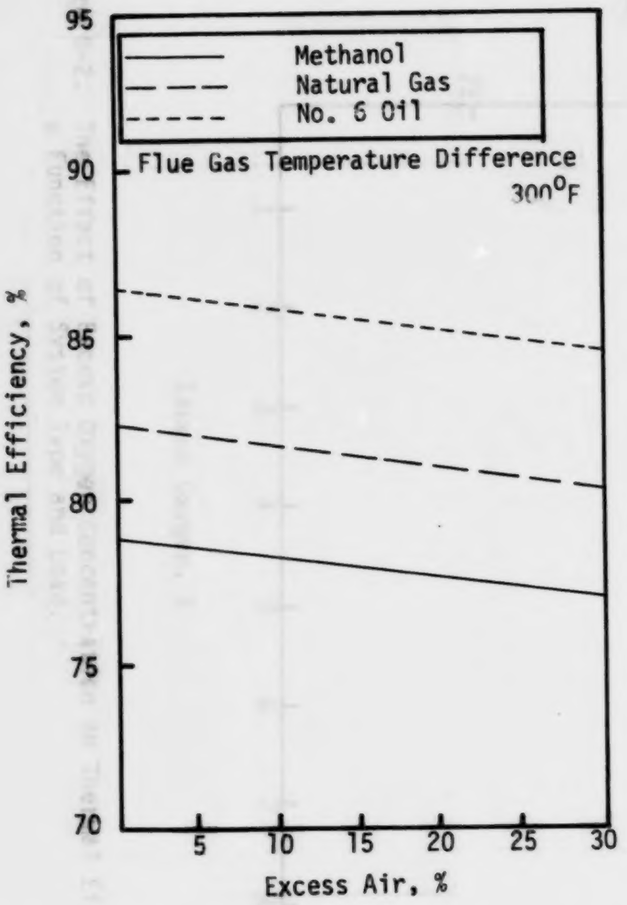


Figure 8-1a. The Effect of Fuel Type on Thermal Efficiency of a Hypothetical Boiler for Constant Flue Gas Temperature Difference.

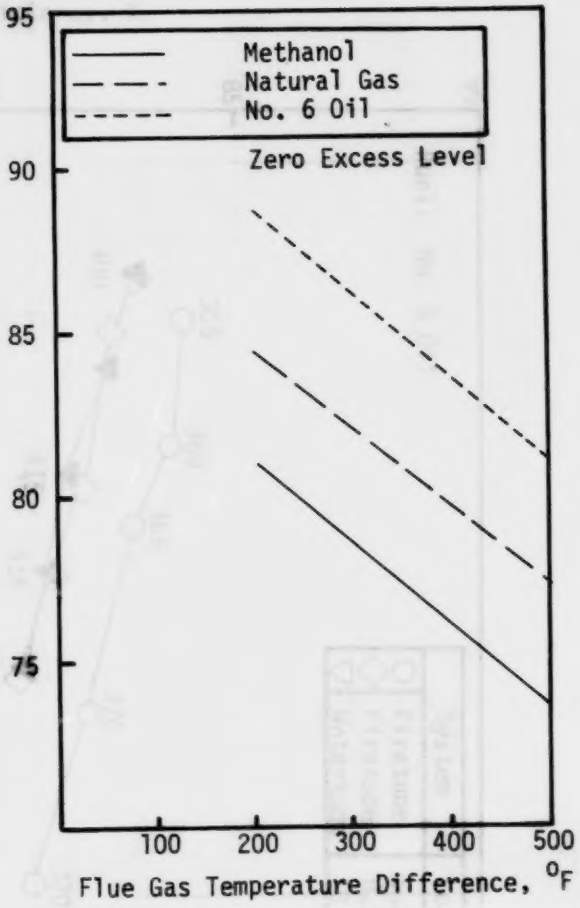


Figure 8-1b. The Effect of Flue Gas Temperature Difference on Thermal Efficiency of a Hypothetical Boiler for Zero Excess Air Level.

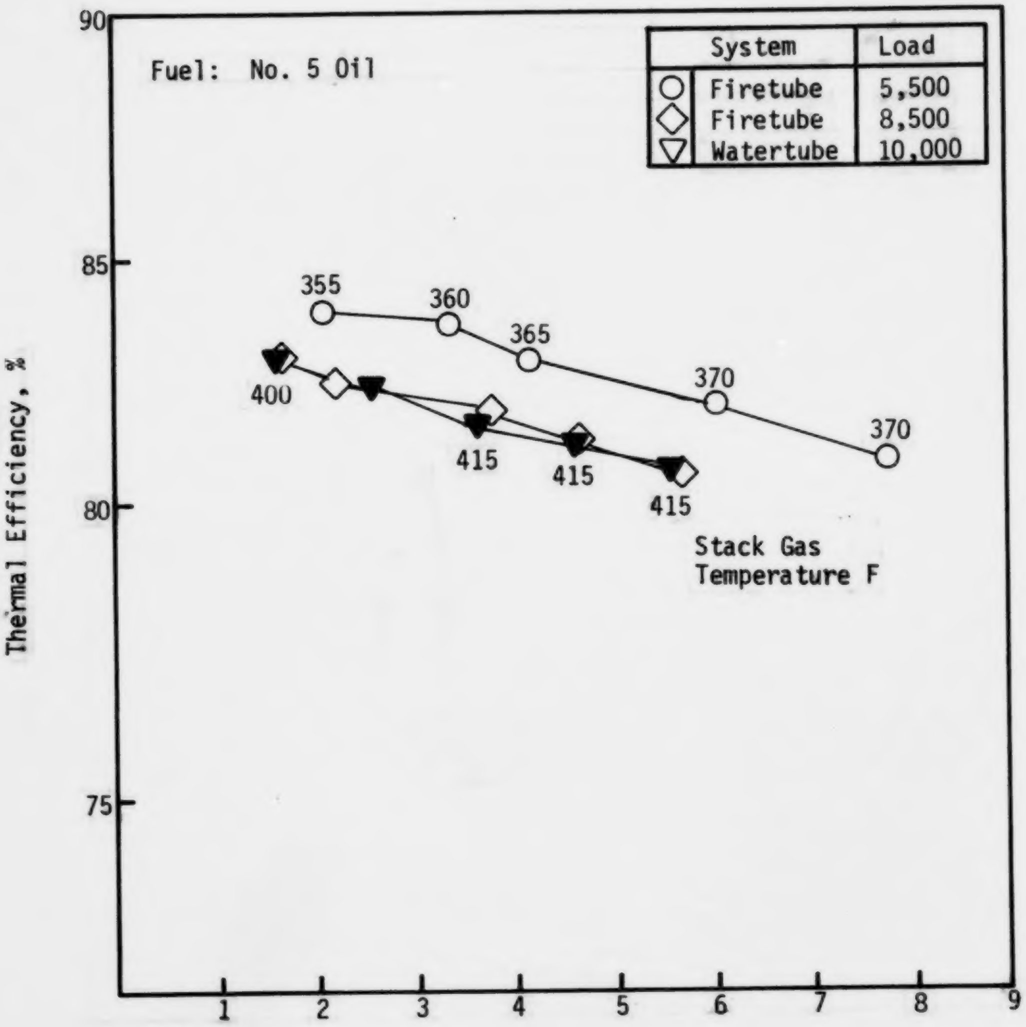


Figure 8-2. The Effect of Excess Oxygen Concentration on Thermal Efficiency As a Function of System Type and Load.

The thermal efficiency of the firetube boiler with methanol firing as a function of excess air level for several loads is displayed in Figure 8-3. It can be seen that the thermal efficiency increases with increasing load at low excess oxygen levels by approximately one percentage point. The relationship between load and thermal efficiency with methanol is inverse to that observed for fuel oil firing. A comparison of thermal efficiency of the firetube boiler firing fuel oil, natural gas, and methanol is presented in Figure 8-4 as a function of excess oxygen level at comparable firing rates. The relative magnitudes of thermal efficiency from the three fuels and their response to excess oxygen concentration is similar to the calculated theoretical thermal efficiency. A similar comparison is made for the watertube boiler in Figure 8-5 and the results are nearly identical to those obtained with the firetube.

The effect of the mass percentage of flue gas recirculation on the thermal efficiency of both the firetube and watertube boiler with the various fuels is illustrated by the results presented in Figures 8-6 to 8-8. The effect of FGR on firetube thermal efficiency can be seen to be almost independent of the amount of FGR. Figures 8-6 compares thermal efficiency with methanol for different loads and excess air levels as a function of the amount of FGR. Thermal efficiency is again noted to be independent of FGR, and only slightly dependent upon load. The data does indicate that a decrease in the excess oxygen level of two percent is responsible for an increase in thermal efficiency of approximately one percentage point. A comparison of the effect of FGR on thermal efficiency of the firetube for No. 5 oil, natural gas, and methanol is given in Figure 8-7.

Similar data for measured thermal efficiencies with the watertube boiler for various fuels as a function of FGR are given in Figures 8-7 and 8-8. Thermal efficiency is noted to be dependent upon the excess oxygen level but almost independent of the FGR rate.

The effect of combustion air staging on measured thermal efficiency for the firetube boiler is displayed in Figueres 8-9 and 8-10. Data for fuel oil firing is given in Figure 8-9 as a function of the percentage of stoichiometric air at the burner exit for two overall excess air levels. Thermal efficiency is independent of the amount of staging for both excess air levels. Data for staging with natural gas is given in Figure 8-10, and also indicates that thermal efficiency is independent of the amount of staging for both high and low excess air levels.

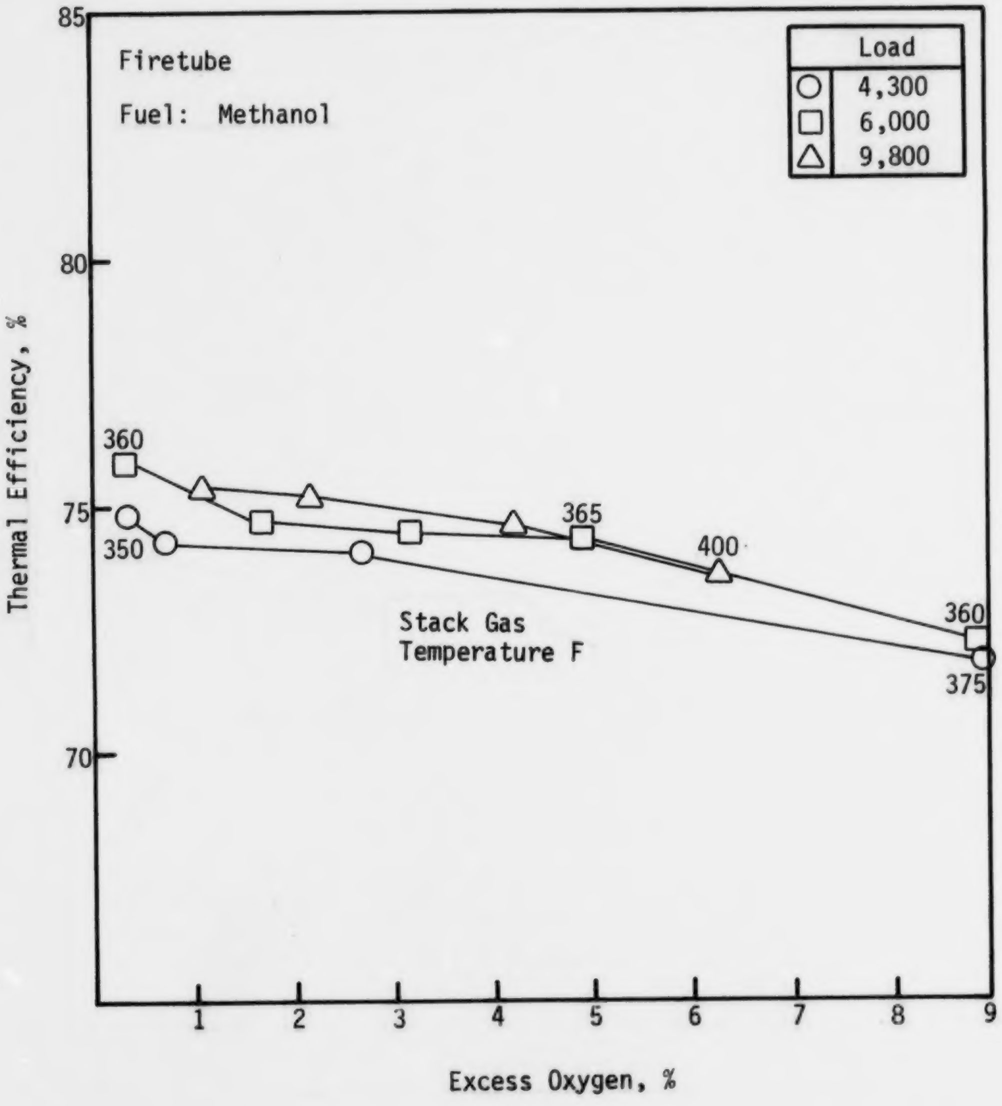


Figure 8-3. The Effect of Excess Oxygen Concentration of Thermal Efficiency for Methanol Firing.

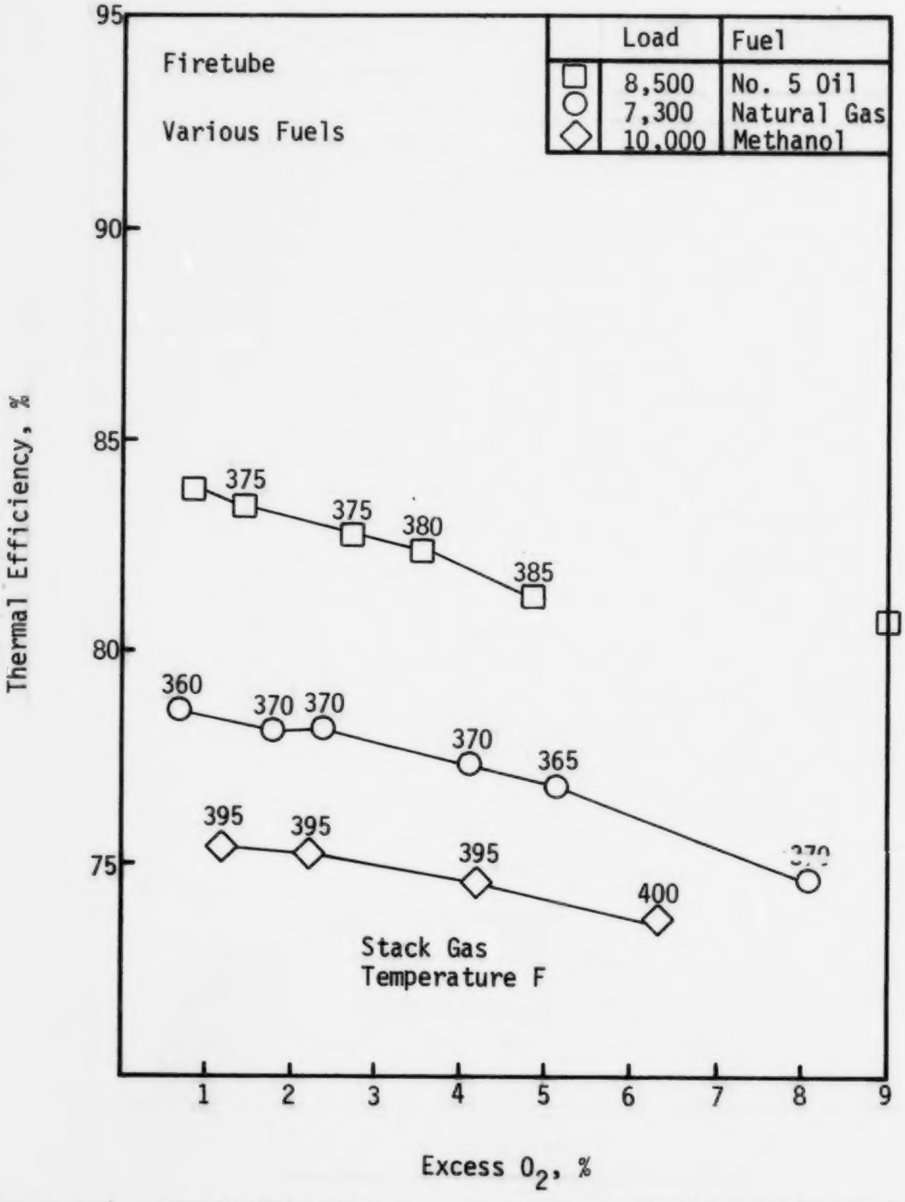


Figure 8-4. The Effect of Excess Oxygen Concentration on Thermal Efficiency from the Firetube for Various Fuels.

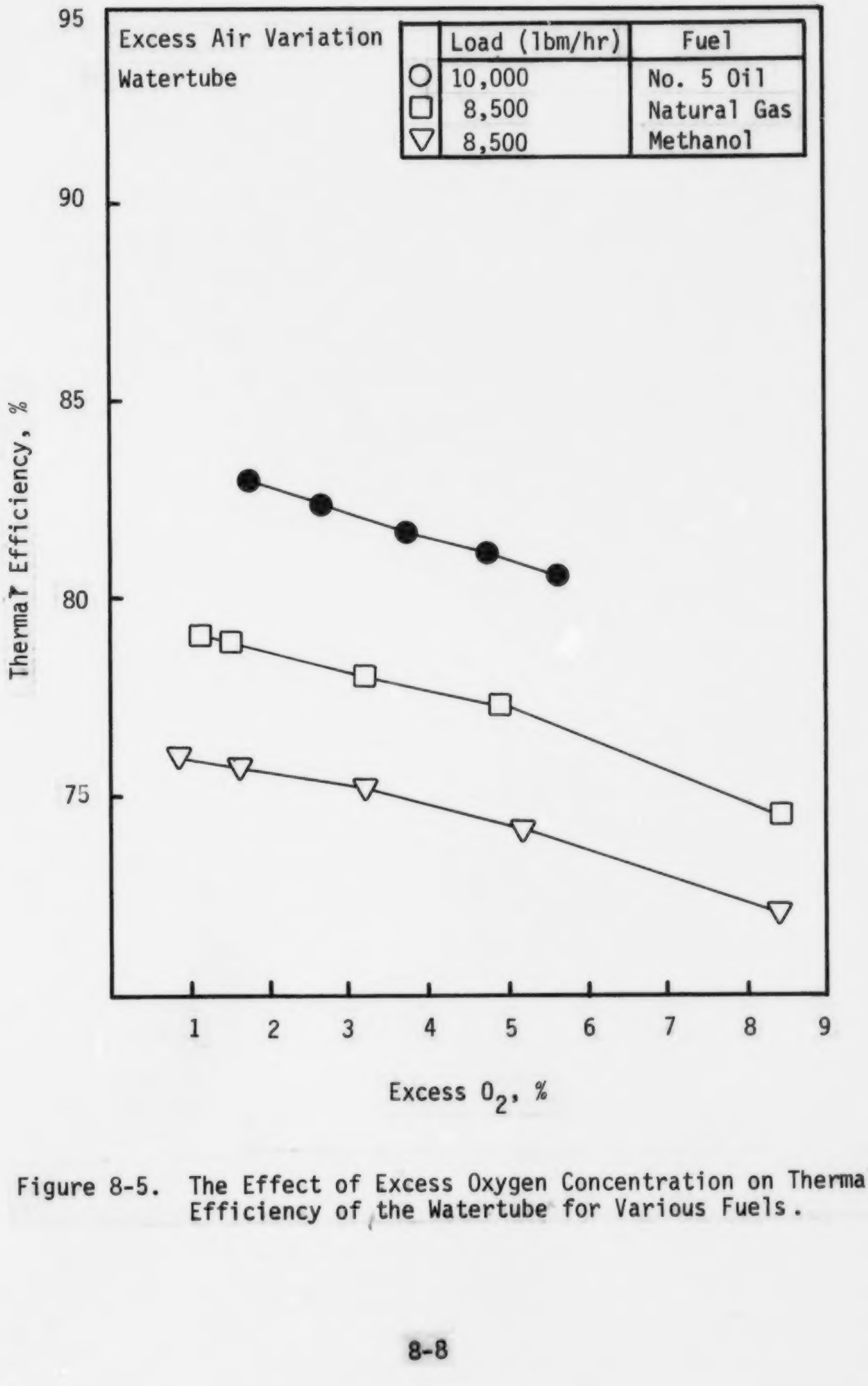


Figure 8-5. The Effect of Excess Oxygen Concentration on Thermal Efficiency of the Watertube for Various Fuels.

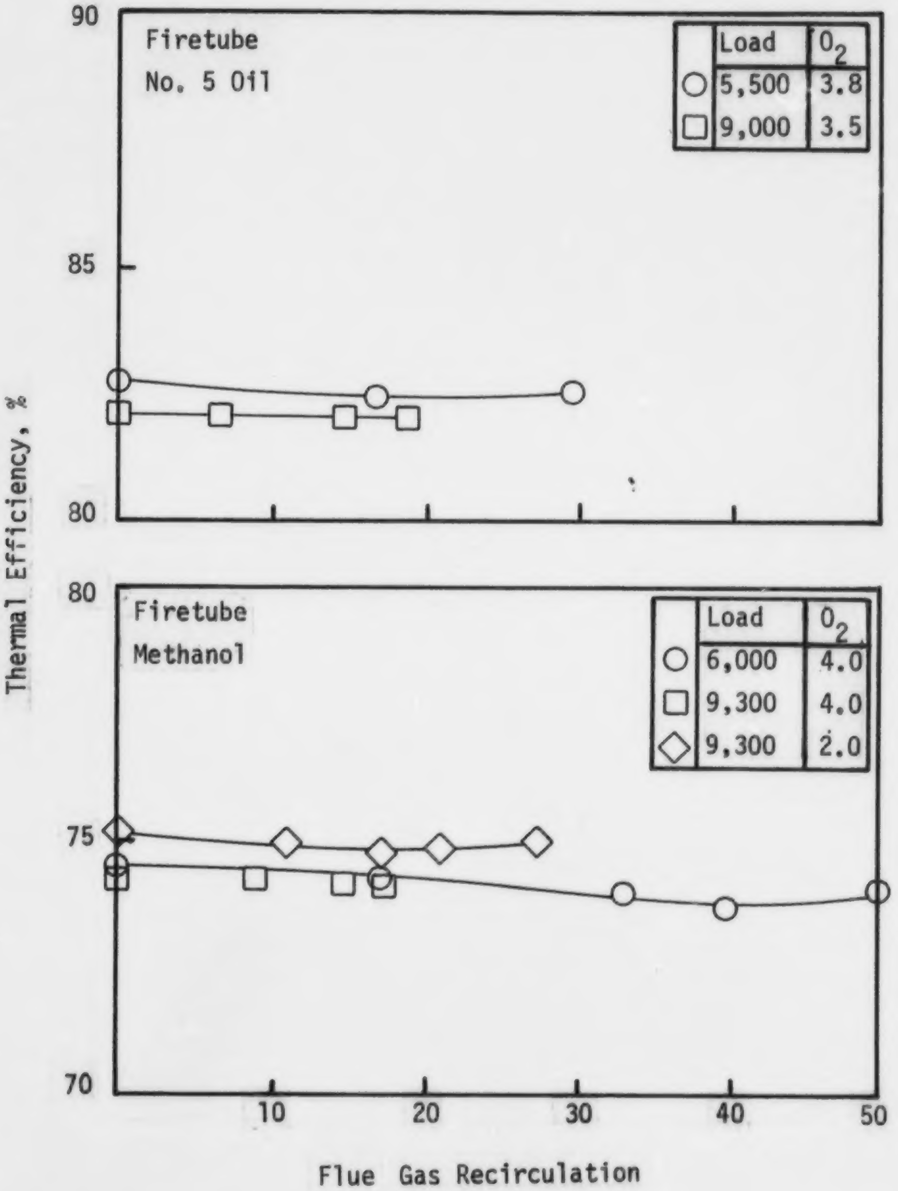


Figure 8-6. The Effect of FGR in Thermal Efficiency from the Firetube For Methanol and No.5 Oil as a Function of Load and Excess Oxygen Concentration.

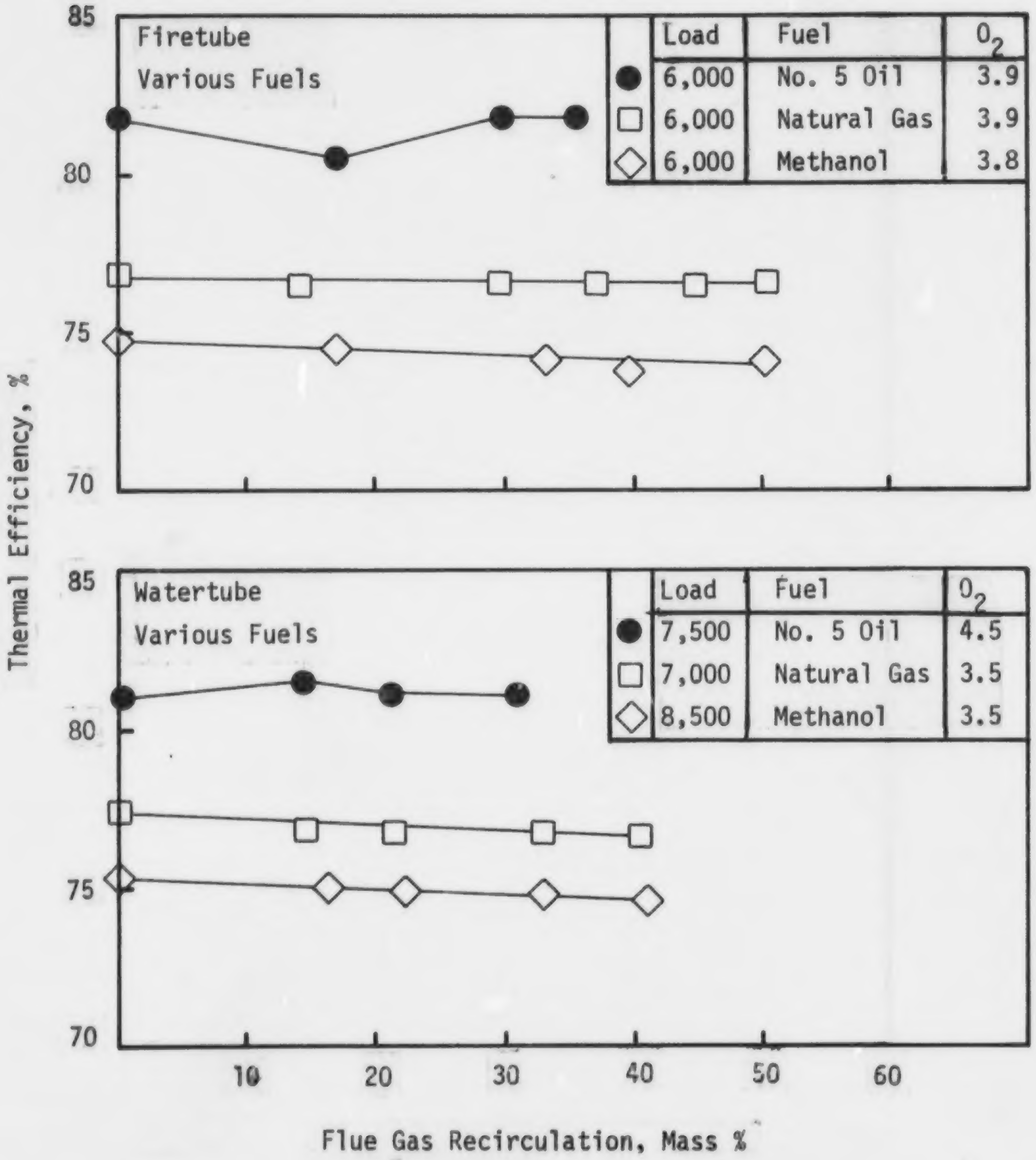


Figure 8-7. The Effect of FGR on Thermal Efficiency of the Firetube and Watertube for Various Fuels.

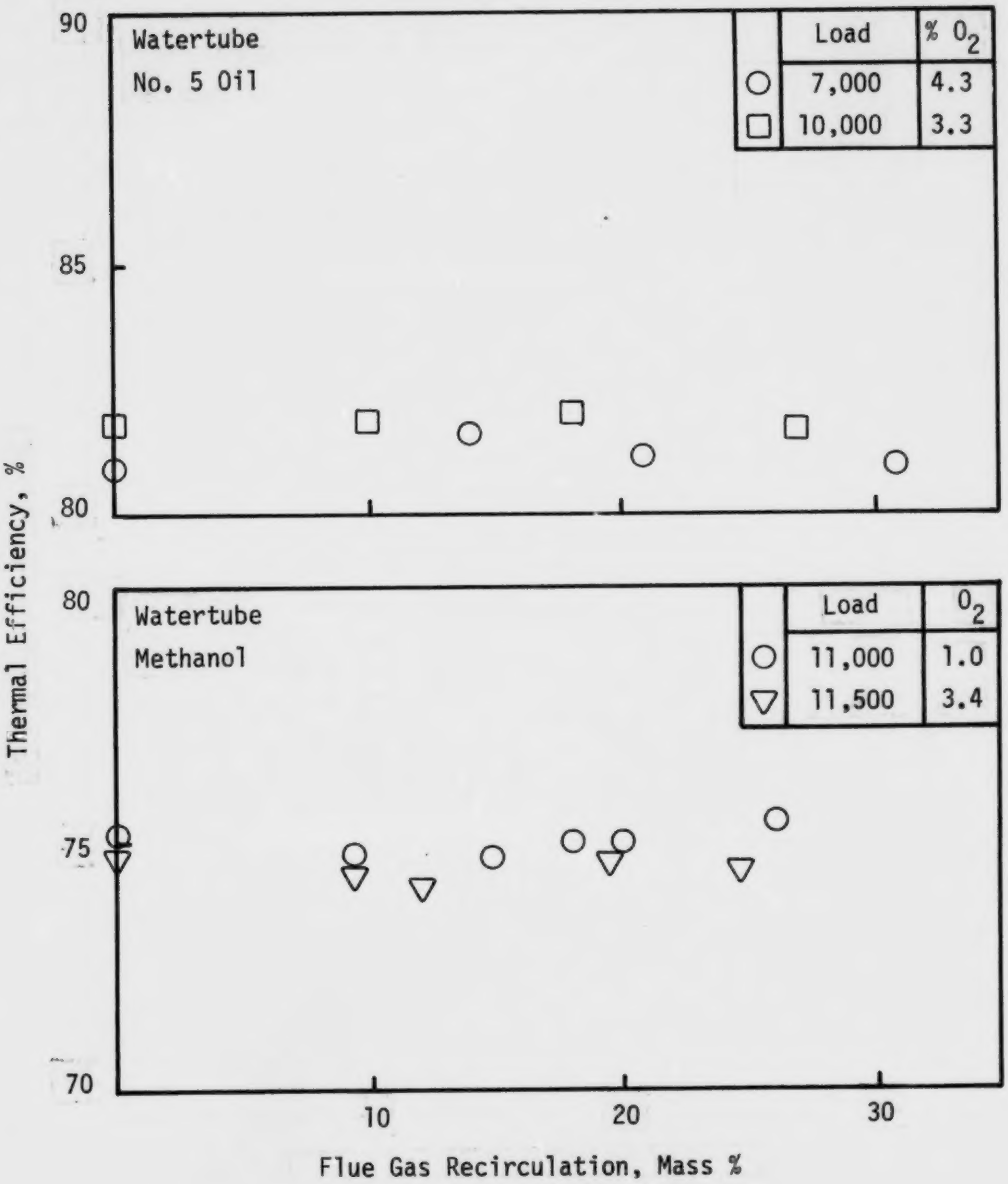


Figure 8-8. The Effect of FGR on Thermal Efficiency of the Watertube for No. 5 Oil and Methanol as a Function of Load and Excess Oxygen.

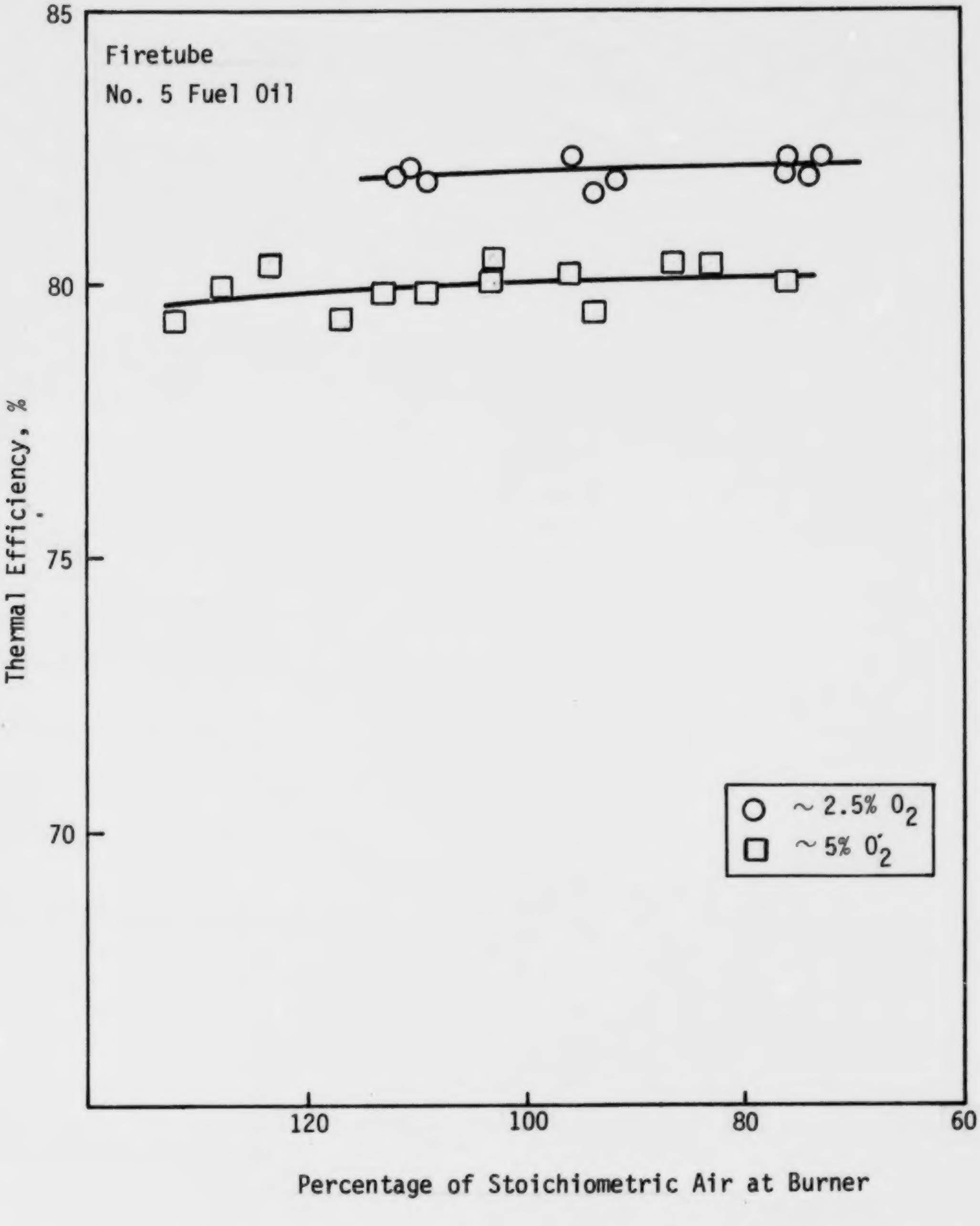


Figure 8-9. The Effect of Burner Stoichiometry on Thermal Efficiency of the Firetube as a Function of Excess Oxygen.

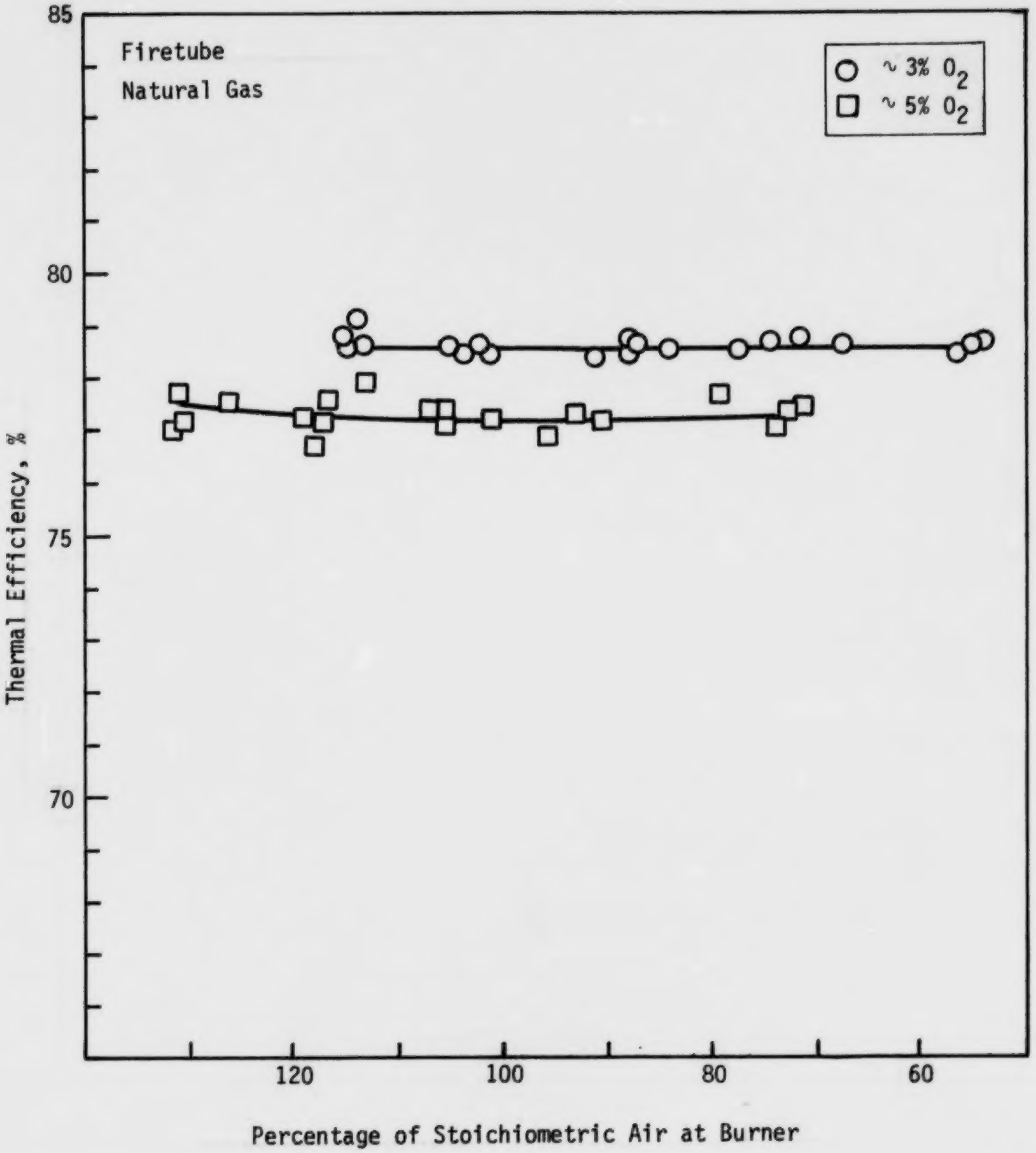


Figure 8-10. The Effect of Burner Stoichiometry on Thermal Efficiency of the Firetube as a Function of Excess Oxygen Level.

8.2 Heat Flux Distribution

Changes in the fuel type or in the heat release rate of the combustion process could have an effect on both the amount and distribution of heat transferred to the combustion chamber walls due to changes in gas temperature distribution and emissivity. If the heat flux distribution is significantly changed such that any area of the heat transfer surface is exposed to a relatively large heat flux, thermal stresses could cause subsequent structural problems. The laboratory firetube, as described previously in Section 4, is designed to allow determinationof the heat flux profile to the heat transfer surfaces. This heat flux profile can be described by both the individual and cumulative temperature rise of the Downtherm coolant in the seven cooling sections.

The individual and cumulative coolant temperature rise for No. 6 and No. 2 fuel oil, natural gas and methanol for the six cylindical coolant jackets and the elbow section are shown in Figure 8-11. Due to the significantly larger surface area of the elbow section in comparison to the cylindrical sections, the temperature rise of the elbow section has been normalized to an equivalent area basis. Comparison of the individual temperature rise for each fuel shows no definite trend, although some general observations can be made. The largest temperature rise for all fuels occurs in one of the first three sections due to the proximity of the flame zone. In the fourth cooling section, the heat flux drops significantly, and is the same for all fuels. The heat flux then increases in either of Section 5 or 6 before decreasing in the last section. Inspection of the cumulative temperature rise indicates that most of the heat is transferred to the firetube combustion section by No. 6 and No. 2 fuel oil, and the least by natural gas. This is not necessarily inconsistent with the previous discussion of the effect of fuel type on thermal efficiency as the measured heat flux is that delivered to the combustion chamber section only.

It is important to determine whether modifications to the combustion process for the control of pollutant emissions can affect the distribution of heat flux at the combustion chamber walls. The effect of FGR on the distribution of heat flux for No. 6 oil, natural gas, and methanol is shown in Figures 8-12 to 8-14. The presence of FGR does not radically change the relative distribution of heat flux for all three fuels. The first coolant section is affected the most, with heat flux absorbed in that section reduced by more than 50 percent for natural gas and methanol and 40 percent for No. 6 fuel oil. The decrease is most likely

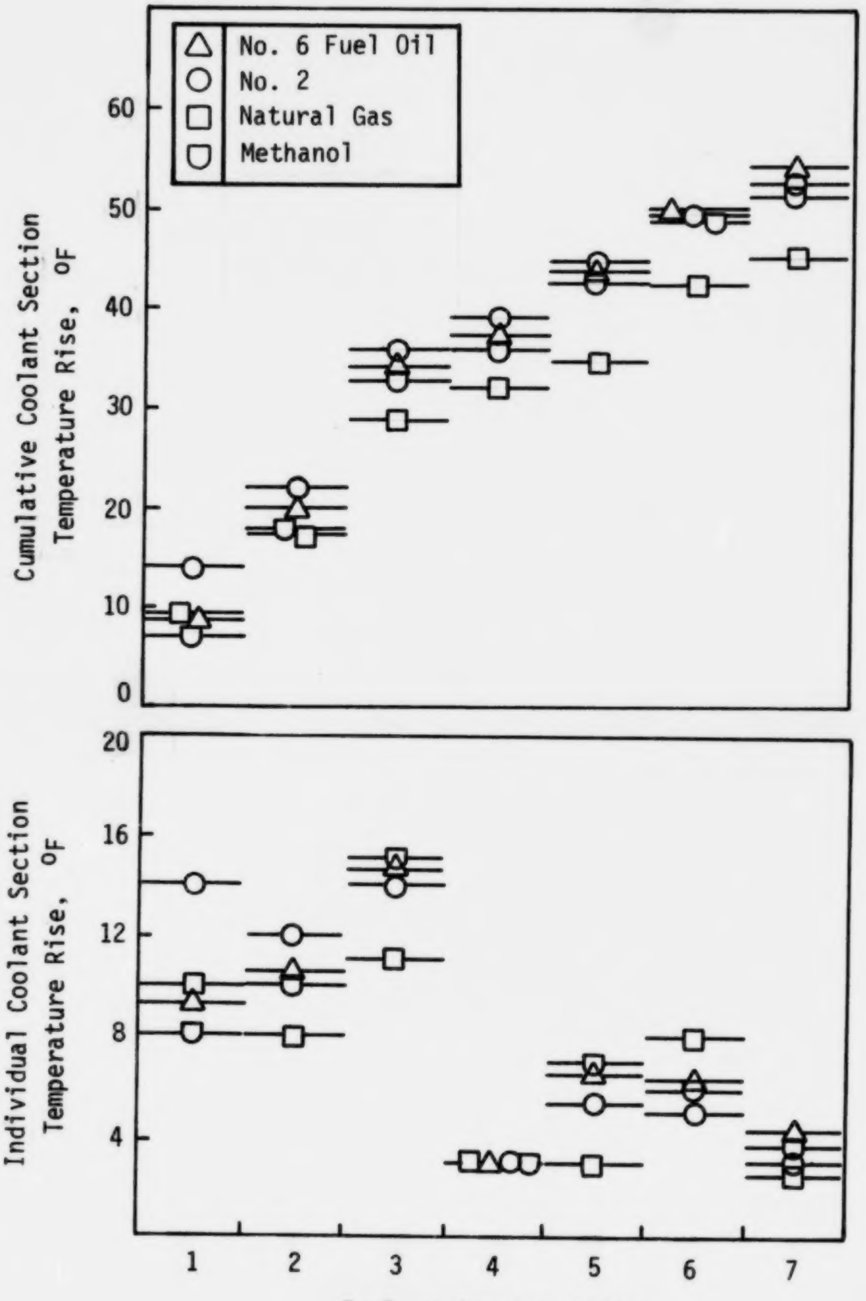


Figure 8-11. Effect of Fuel Type on Individual and Cumulative Coolant Section Temperature Rise.

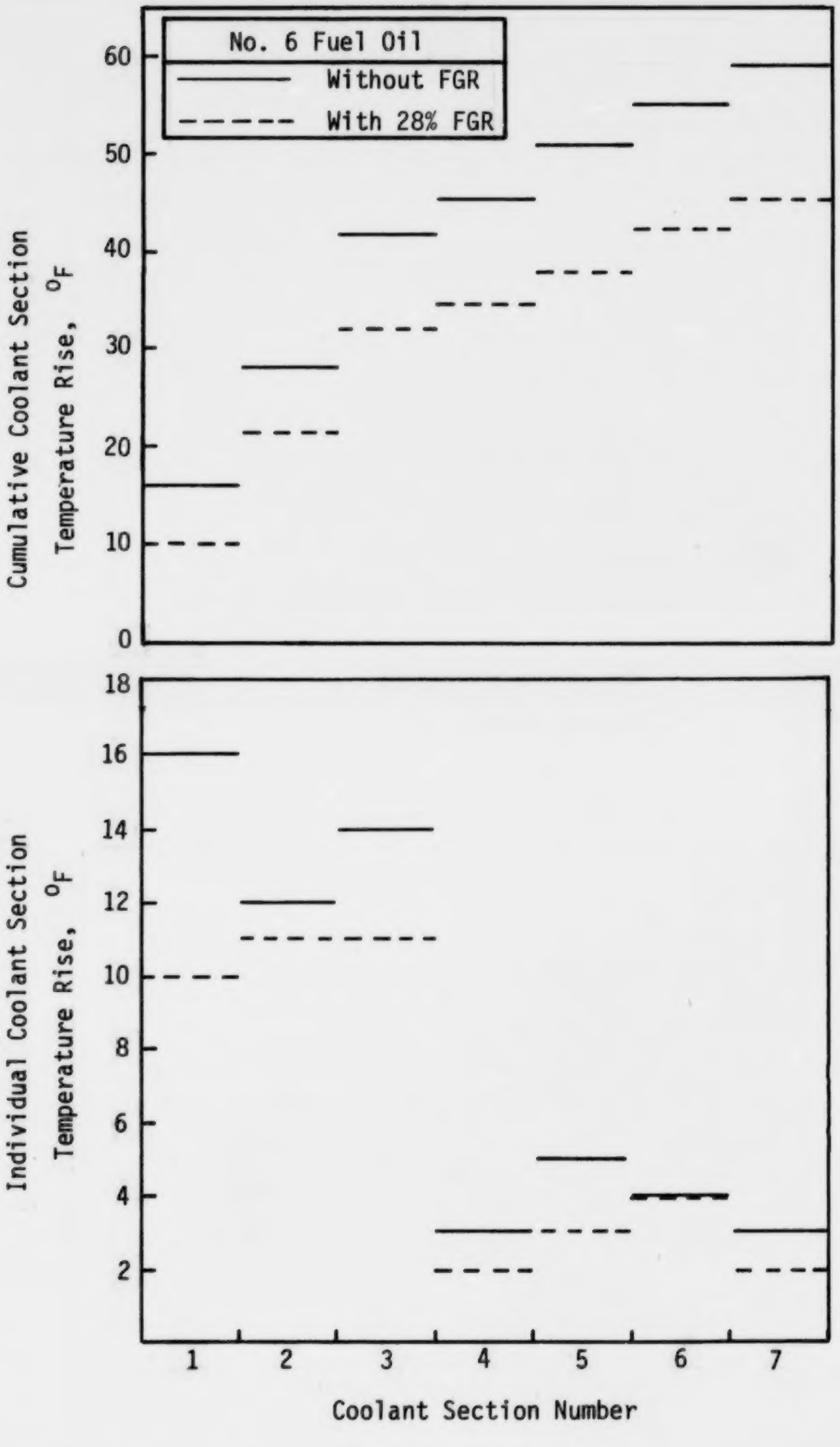


Figure 8-12. Effect of FGR on Individual and Cumulative Coolant Section Temperature Rise with No. 6 Oil.

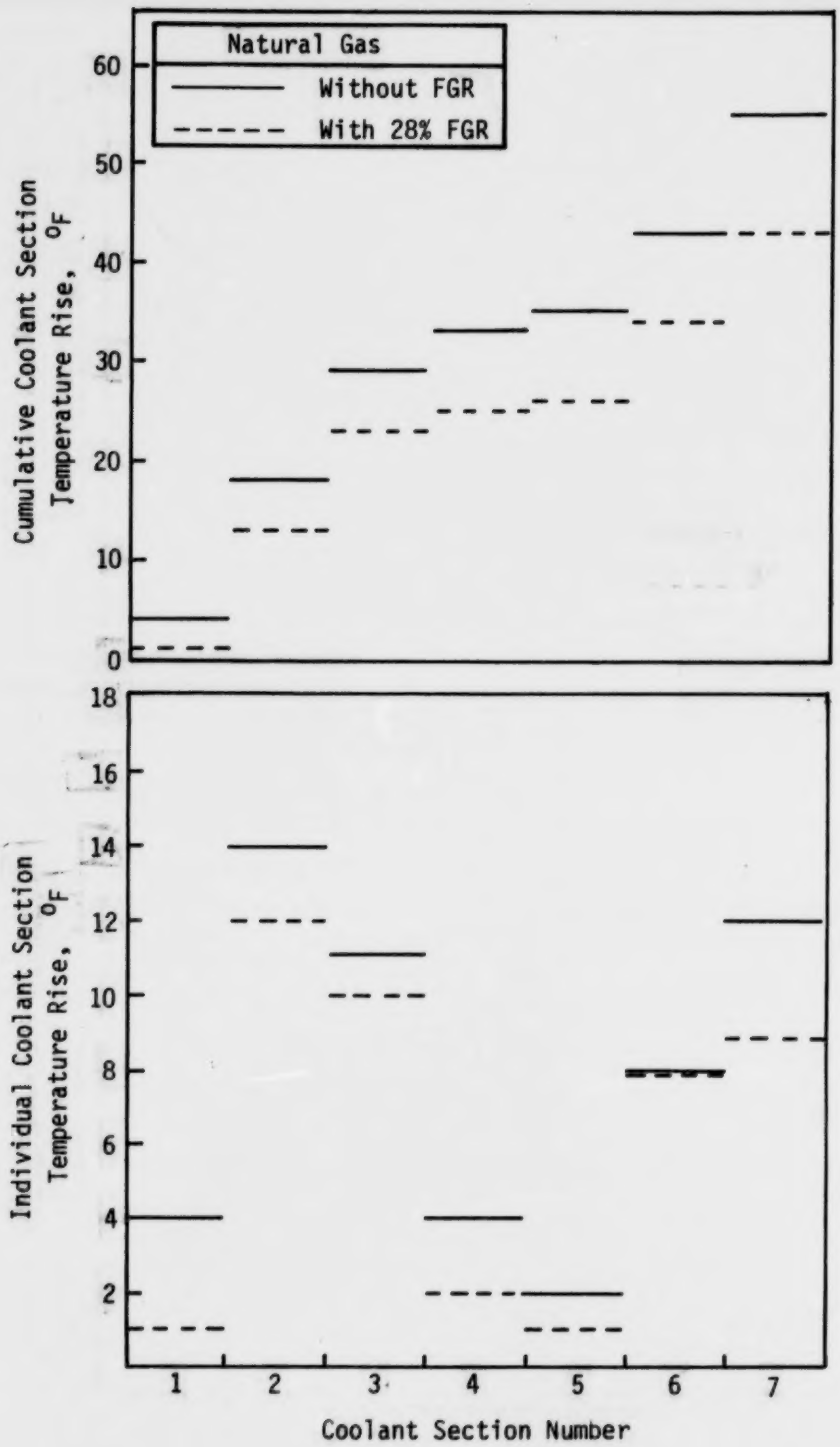


Figure 8-13. Effect of FGR on Individual and Cumulative Coolant Section Temperature Rise with Natural Gas.

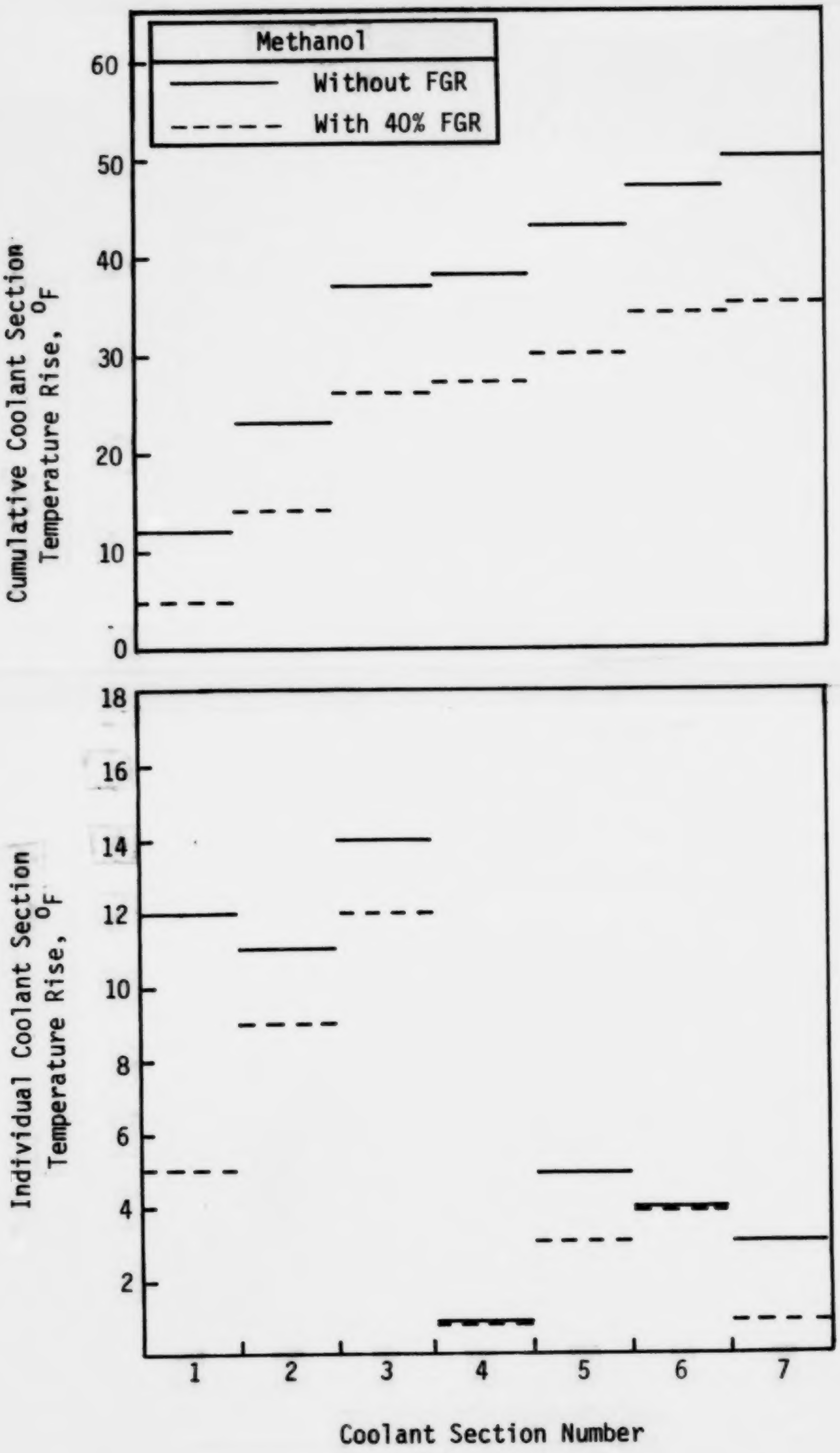


Figure 8-14. Effect of FGR on Individual and Cumulative Section Temperature Rise with Methanol.

due to the lower flame temperatures associated with the diluated fuel/air mixture. The relative distribution of heat flux does not change significantly for all fuels, the the magnitude of heat transferred to each section is less. The total heat transfer to all sections is less for all fuels as indicated by the cumulative temperature rise of the coolant. This decrease in heat transfer to the firetube section would mean an increase in heat loading in the convective section. Whether the overall efficiency would change would be dependent on the amount of increase in heat loading and the design of the convective section. Results from the field test indicate that this is unlikely to be an important practical consideration.

The effect of combustion air staging on the distribution and amount of heat absorbed in the firetube section is shown in Figures 8-15 and 8-16. As with FGR, the relative heat flux distribution does not change radically, but unlike FGR, the effect of staging can act to increase or decrease the heat transfer to certain coolant sections. The largest decrease in heat transfer occurs in the first three sections, but is not as large as that incurred with FGR. This is probably due to the lower bulk temperatures of the gases associated with fuel-rich combustion. The heat transfer to the last two sections is higher under staged conditions due to the delayed heat release, and consequently, higher bulk gas temperatures in that region of the firetube. The total heat transferred to the firetube section as indicated by the cumulative temperature rise is less with staging for both fuels; the decrease is not as large as with FGR.

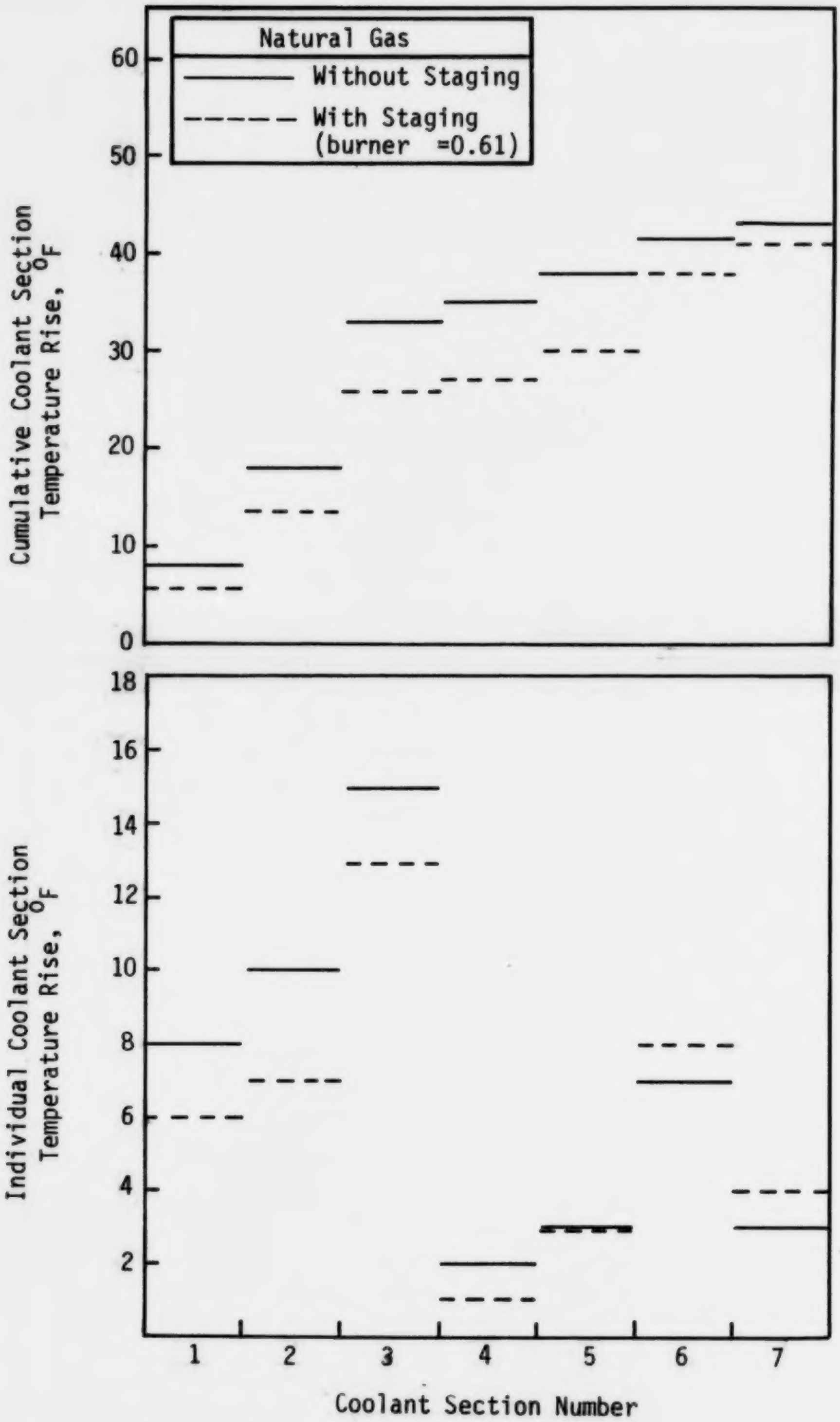


Figure 8-15. Effect of Combustion Air Staging on Individual and Cumulative Coolant Section Temperature Rise with Natural Gas Firing.

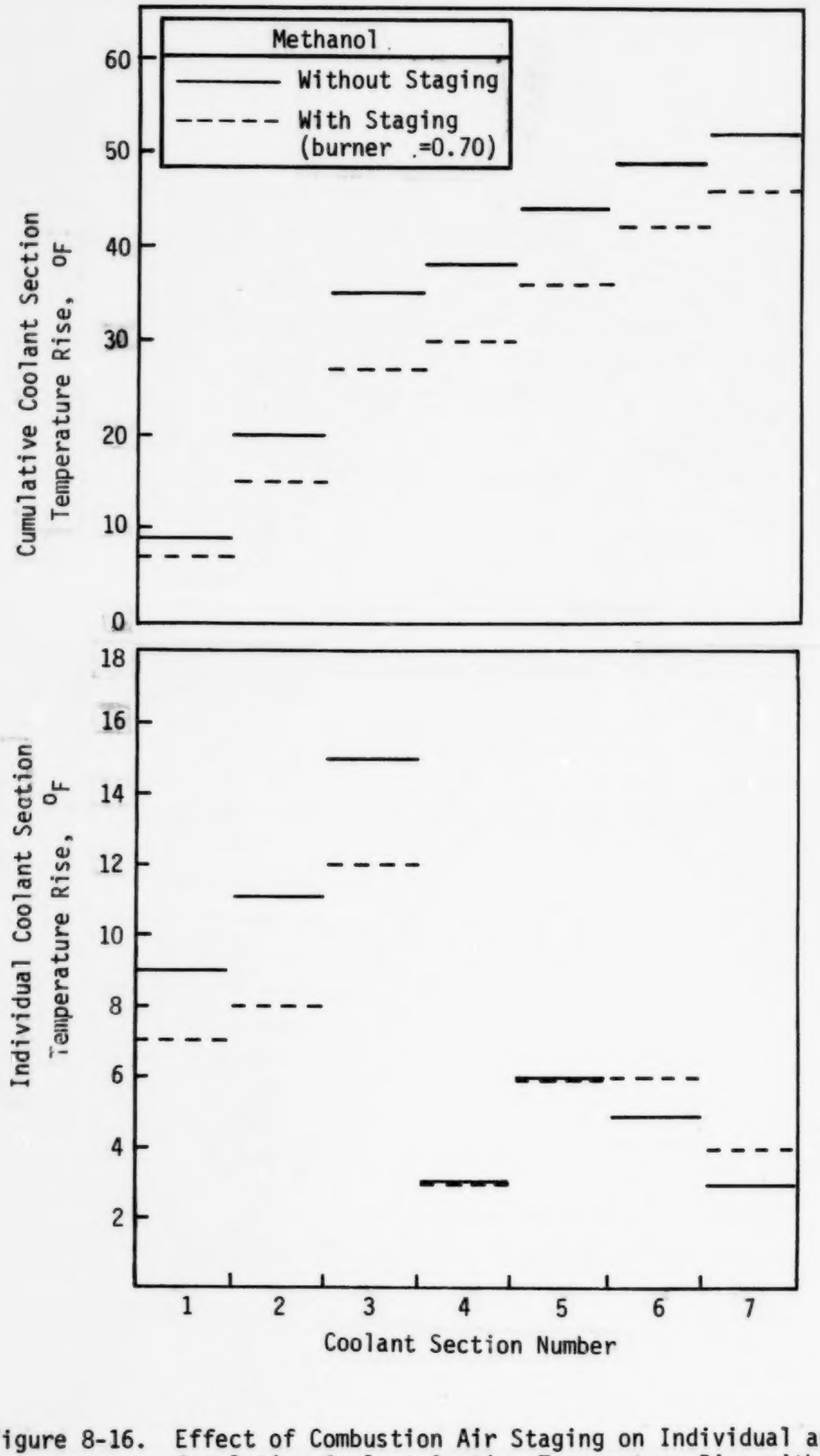


Figure 8-16. Effect of Combustion Air Staging on Individual and Cumulative Coolant Section Temperature Rise with Methanol.

9.0 PRACTICAL IMPLICATIONS AND RECOMMENDATIONS

This report describes the results of one of several investigations sponsored by the Environmental Protection Agency demonstrating the applicating of control techniques to reduce the amount of nitrogen oxides emitted by stationary combustion sources. The intent of these techniques is to reduce the amount of pollutant formation rather than to remove the pollutant from the effluent gases. This report concentrates upon one class of stationary source - single burner package boilers. The control techniques demonstrated include operational changes as well as hardware modifications to recycle combustion products or to stage the combustion process. With the exception of studies involving methanol, complete originality cannot be claimed for any of the work described in the preceding sections. Therefore, it is pertinent to consider what is the present state-of-the-art for the control of NO_x emissions from package boilers.

The ultimate objectives of the type of work described in this report are the collection of an authenticated data base concerning the application of a particular control technique and the provision of information which will aid in the design of commercial equipment which might have to be constructed to meet emission standards. Thus, to satisfy the second objective, investigations such as this one should rank the available control options and provide design criteria which could be applied by industry.

The control of pollutant emissions should not supersede concern for energy conservation. Thus, a primary factor in the choice of the most appropriate emission control technique for package boilers is its impact upon thermal efficiency. This does not imply that capital cost nor attendant increases in auxiliary plant power requirements can be ignored. However, if the application of some form of pollutant control results in fuel savings, then this saving can be offset against other expenses. Therefore, in the field tests discussed in this report, both pollutant emissions and boiler efficiency have been determined.

Four control options have been investigated and compared on the basis of the tradeoff between changes in NO_x emission and their effect upon thermal efficiency:

1. Reduced excess air firing which may include the necessity for burner tune-up;
2. Fuel changes;

3. Flue gas circulation; and
4. Staged combustion.

Figure 9-1 presents a plot of NO_x emissions (ppm corrected to 0% O_2) against thermal efficiency which compares the influence of these various techniques for the firetube boiler. The data plotted on this graph is representative of the results discussed earlier, but not all-inclusive. The CO limit shown represents the condition at which the CO concentration shows a marked increased from the baseline value. It can be seen that:

1. Reduced excess air firing will result in a decrease in both NO_x emissions and an increase in thermal efficiency. While no hardware modifications are required, this technique can only be considered as a control method for those boilers which are not operated at conditions giving maximum thermal efficiency unless burner modifications can result in even greater reductions in excess air levels.
2. Neglecting economic considerations, NO_x emissions would be reduced if methanol were to be used in preference to either natural gas or fuel oil in unmodified boilers. However, this decrease in NO_x emissions would also be accompanied by a decrease in thermal efficiency.
3. Flue gas recirculation does not have a strong influence on thermal efficiency (within the accuracy of the measurements, efficiency was unchanged with the addition of FGR), however, auxiliary power requirements increase. As a control technique it is most effective for gaseous and liquid fuels which do not contain bound nitrogen.
4. No influence of staged combustion on thermal efficiency was observed during the staging experiments. As indicated in Section 8, the staging equipment used in the firetube boiler was not nearly as effective with natural gas as with fuel oil.

The numercial values presented in Figure 9-1 relate to one boiler; however, the general characteristics and the observed trends will be similar for all boilers of this size. Due to the severe penalty in thermal efficiency, the use of methanol as a control technique to attain very low NO_x emissions can be discounted unless methanol costs are comparable to those of conventional fuels. Thus, significant reductions in NO_x emissions with increased thermal efficiency can only be achieved by combined application of staged combustion and/or flue gas

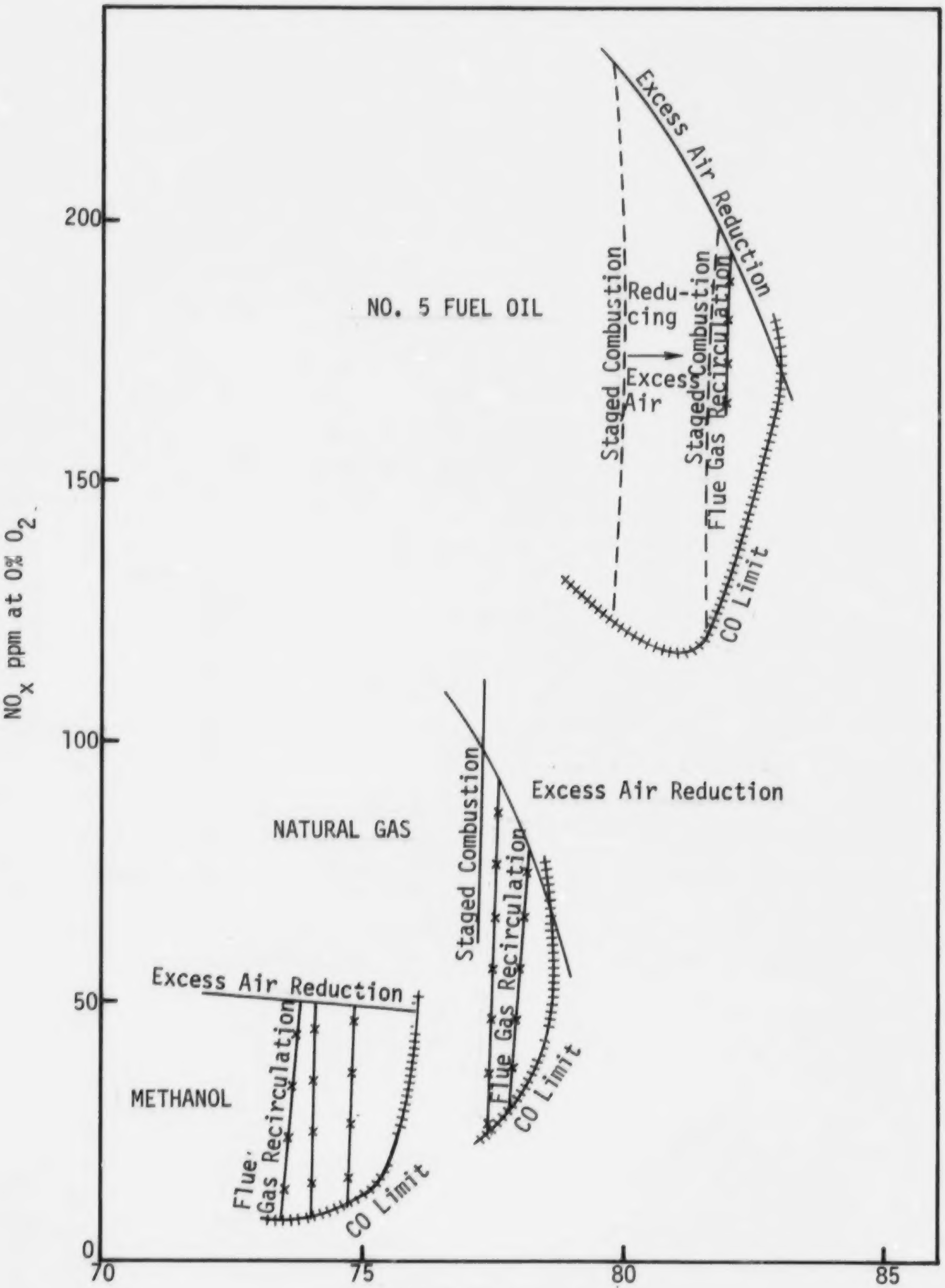


Figure 9-1. NO$_x$ Versus Thermal Efficiency for the Firetube Boiler Field Tests.

recirculation with low excess air firing. The results presented in Figure 9-1 indicate two problems which require further consideration.

1. The use of staged combustion or flue gas recirculation optimized for maximum NO_x reduction, with one particular fuel in one boiler may not give optimum results with another fuel.
2. Staged combustion appears to be the better long-term approach, although reductions in NO_x emissions were limited to 50 percent with fuel oil firing and increased NO_x emissions were observed at some operating conditions with natural gas.

Fundamental considerations indicate that flue gas recirculation will not provide adequate control possibilities for those fuels containing significant amounts of bound nitrogen. However, for natural gas, NO_x emission can be reduced to very low levels (less than 20 ppm) without attendant design problems. At present the major uncertainties associated with the application of flue gas recirculation concern long-term maintenance and reliability and the appropriate control equipment. The requirement of dual fuel capability for most industrial equipment and the decreased availability of natural gas necessitates the development of other acceptable control techniques for industrial boilers. From a cost-effectiveness viewpoint, it is necessary that alternative control techniques should be developed which are equally effective with natural gas and residual fuel oils.

The potential of staged combustion techniques to control both fuel and thermal NO_x emissions has been appreciated for some time. Considerable success has been reported for utility boilers when the staging process was achieved by biased burner operation or off-stoichiometric firing. However, neither the results presented earlier nor those described by Cato et al can be considered to be more than promising for package boilers, particularly when compared to the results reported by Siegmund and Turner(21).

The most discouraging feature of all the attempts to stage the combustion process in package boilers has been the lack of generality, and therefore, applicability of the results. Also, there can be serious questions raised as to the practicality of the methods of staged air injection. The staging equipment used in the firetube boiler and described earlier in this report was considered as "experimental", and although it withstood the environment within the firetube during the staging investigations, it could not be considered as a representative design. Cato et al inserted staged air injectors from the front wall of one watertube boiler and used sidewall injectors with another boiler. The sidewall injection system was much more effective in reducing NO_x emissions from both fuel oil and natural gas than the front wall system. This could be attributed to differences in the burner and/or boiler design or the fact that the most effective system was used with preheated combustion air. With natural gas firing and the front wall staged air injection system, NO_x emissions initially increased as the burner stoichiometry was decreased and with 70 percent of the stoichiometric air through the burner were essentially the same as the unstaged emissions.

The possibility of increasing NO_x emissions from gas-fired systems by attempting to stage the combustion process have been described before. Heap et al(19) described early results using the burner as a staging device in which emissions from a natural gas flame were increased by a factor of three when all the combustion air was not supplied through the primary burner throat. Recent laboratory experiments carried out by De Soete(20) in premixed stratified systems indicate that staging could cause an increase in NO_x emissions. Therefore, the results of Cato et al and those described in Section 7 of this report are not with precedent.

There are two possible explanations for the failure of certain staging systems with natural gas firing. The rate of thermal NO formation is strongly dependent upon temperature and in the systems under consideration, both reaction zone temperatures and the rate of temperature decay of products of combustion from flame temperatures to bulk gas temperatures are strongly dependent upon mass transfer processes. The reduction in burner momentum due to the reduction in the burner air flow will affect the rate of heat release in the primary zone. If significant heat loss does not take place before the introduction of the staged air, higher reaction zone temperatures or reduced quench rates may be found in the staged condition than in the unstaged condition. A second possibility is associated with the formation of nitrogen containing radicals (e.g., CH or NH_2) in the rich primary zone which are then converted to NO in the secondary zone.

For oil-fired systems, it is apparent that although considerable experimental data are available, no generalized design procedure has been developed for staged combustion systems. Once more, the major problem is developing a design which will be acceptable to the process and yet, reduce NO_x emissions. Techniques which require excessive reduction in boiler capacity are of limited value. Some time ago, Heap et al(22) described burner designs which gave low NO_x emissions for fuels containing bound nitrogen (pulverized coal and No. 6 oil) by restricting the rate of fuel/air mixing. Unfortunately, these designs also produced long flames which may not be suitable for package watertube boilers, but could perhaps be utilized in firetube boilers.

The major dilemma uncovered by the results presented in this report is associated with dual fuel or residual fuel-fired boilers. If methanol production from coal were to be economically attractive, then conversion to methanol firing would introduce few practical problems and it would certainly reduce NO_x emissions from package boilers. However, conversion to alcohol fuels would have to be given national priority and it is not a short-term solution. For natural gas-fueled boilers, flue gas recirculation is extremely effective but the only practical solution for dual fuel-fired boilers could be a combination of flue gas recirculation and staged combustion control. The major disadvantage of staged combustion as a technique for NO_x control for package boilers appears to evolve from a lack of understanding of the basic phenomena which precludes the production of an effective design.

The general requirements of a staged combustion system for residual oil-fired equipment have never been defined beyond a requirement for fuel-rich primary zone followed by a secondary burnout zone. The optimum temperature, time and percent of theoretical air in the primary zone have never been identified. The possibility of carbon formation with the attendant problems of burnout and nitrogen species absorption suggest that the primary zone should operate as close to stoichiometric proportions as possible. It has been suggested that the nitrogen-containing fractions in residual fuel oils have a high boiling point, thus the temperature and residence time in the primary zone are critical to minimizing carryover of nitrogen fragments into the fuel-lean secondary zone. Should the information discussed above be available, it is still not sufficient to allow a combustion system to be designed since the fuel/air contacting process must be managed in such a way as to provide the optimum conditions in the primary zone.

An empirical approach to the problem of oil burner design for firetube boilers has been taken in another EPA-sponsored program (Contract 68-02-1500). Fuel/air contacting are dependent upon burner design, therefore the provision of optimum fuel-rich first stage conditions relay on the fuel spray and air flow distribution. This study has also emphasized that the limitations of the empirical approach are associated with increases smoke emissions before reductions in emission levels greater than 50 percent can be achieved.

If optimum combustion systems for residual oil-fired package boilers are to be designed, it would appear that the most effective program should first return to laboratory situations to learn how NO_x is produced in residual oil flames. Armed with this information, it might then be possible to define those conditions leading to minimum emissions in practical systems.

10.0 BURNER OPTIMIZATION FOR NO_x CONTROL

The previous section discussed the implications of field tests of two package boilers. It was concluded that generalization of the data was a major deficiency together with the moderate success of staged combustion when applied to dual fuel systems. Two problems were identified:

1. The rudimentary information available on the basic phenomena associated with staged combustion of liquid fuels.
2. Practical considerations indicate that most cost effective NO_x control options for package boilers will be associated with the burner system. If the burner can be optimized, package boilers could be retrofitted in the field and new boilers could be built without the need to redesign the basic boiler system.

This section discusses the mechanisms of NO_x formation and the implication for low NO_x burner design.

10.1 Fundamentals of NO_x Formation

It is generally acknowledged that NO_x can be formed by two chemical mechanisms: the high temperature, thermal fixation of atmospheric nitrogen (leading to thermal NO_x), and the oxidation of nitrogen chemically bound in the fuel (leading to fuel NO_x). The formation of thermal NO_x is extremely sensitive to flame temperature; however, flame temperature has only a relatively small effect on fuel NO_x formation. Consequently, it is important to establish the source of the NO_x produced in flames because of the impact on potential control strategies. This also applies to the source of nitrogen for fuel NO_x formation.

The formation of thermal NO_x was originally believed to occur according to the "Zeldovich mechanism"[23]:

$$M + O_2 \rightleftarrows 2O + M \tag{1}$$

$$O + N_2 \rightleftarrows NO + N \tag{2}$$

$$N + O_2 \rightleftarrows NO + O \tag{3}$$

where M represents other species present in the combusting mixture. The following reaction was added for fuel-rich conditions where substantial amounts of OH may be present:

$$N + OH \rightleftarrows NO + H \tag{4}$$

This "Extended Zeldovich" mechanism was then used to predict NO_x emissions from actual combustion system by assuming that H, OH, and O were in thermodynamic equilibrium prior to the onset of NO_x formation.(24-27) The consensus of this work is that the extended Zeldovich mechanism shown above adequately describes NO_x formation in the postflame zone region; however, there are substantial concentrations of "prompt NO" formed very rapidly in the flame front of gas/air flames and which cannot be rationalized on the basis of this mechanism, assuming equilibrium radical concentrations. It has been suggested(30-34) that prompt NO is the consequence of nonequilibrium radical concentrations in the vicinity of the combustion zone. Significant support for this hypothesis was gained when several researchers, including Bowman(35), Malte(36), and Mitchell(37), experimentally observed "super equilibrium" free radical concentrations in close proximity to the flame zone. These high radical concentrations were believed to be related to chain-branching hydrocarbon reactions of the following types:

$$CH_i + O \rightarrow CH_{i-1} + OH \tag{5}$$

$$CH_i + OH \rightarrow CH_{i-1}O + H \tag{6}$$

Alternately, Fenimore(38) postulated that prompt NO was the result of the reaction of molecular nitrogen with hydrocarbon radicals to form bound-nitrogen radicals which are then oxidized to form NO. For example:

$$CH + N_2 \rightarrow CN + NH \tag{7}$$

In natural gas flames, all of the exhaust NO_x emissions are the results of thermal fixation; however, in liquid fuel combustion, a portion of the exhaust NO_x may be the result of the oxidation of nitrogen chemically bound in the fuel. In heavy oils, the fuel nitrogen exists primarily in large, multi-ring organic compounds. The detailed fate of these molecules during combustion has not been fully resolved, but the available data suggest that, in general, the nitrogen behavior closely parallels that of carbon: a portion of the nitrogen is evolved with the light hydrocarbon volatiles and the rest may remain in the char matrix and is subsequently oxidized heterogeneously.

Once the volatile nitrogen species reach the free-stream, they may be either oxidized to NO or ultimately reduced to N_2. As shown in Table 10-1, a number of experimental studies have been conducted on the addition of typical nitrogenous compounds to flat, laboratory flames. The data in Table 10-1 indicate that fuel nitrogen is a large potential source of NO emissions and

Table 10-1. Studies on the Addition of Nitrogenous Compounds to Flat, Laboratory Flames

Investigator	Fuel	Nitrogen Compounds Investigated
Shaw(41)	CO	Pyridines, amines, cyanides
Wendt(42)	CH_4	NH_3
Axworthy(39)	CH_4	HCN, NH_3, NO
DeSoete(40)	CH_4, C_2H_4	C_2N_2, NH_3, NO
Sarofim(44)	CH_4	NH_3, NO, pyridine, methylamine
Sawyer(45)	C_3H_8	C_2H_2, NH_3, NO
Haynes(46)	C_2H_2, C_2H_4, C_3H_8, CH_4, H_2	CN species, NO, NH
Merryman(47)	CH_4	Methylamine, pyridine, piperidine

that the oxidation of many nitrogen-containing compounds to NO is rapid, occurring on a time-scale comparable to that of the combustion reactions.(35) For example, Axworthy(39) found that above a stoichiometric ratio of 0.8, more than 50 percent of the inlet nitrogen appeared as NO in the exhaust.

The fundamental mechanism controlling the oxidation and reduction of the volatile nitrogen species is believed to be:

$$\left\{\begin{array}{l}\text{volatile}\\ \\ \text{fuel N}\end{array}\right\} \rightarrow \text{HCN} \rightarrow \left\{\begin{array}{l}NH_3\\ NH_2\\ NH\end{array}\right\} \begin{array}{l}\nearrow \text{NO (oxidizing)} \quad (8)\\ \\ \searrow N_2 \text{ (reducing)} \quad (9)\end{array}$$

Under oxidizing conditions, the NH_i species are directly oxidized to form NO. Under reducing conditions, N_2 formation dominates probably according to the following raction(40):

$$NH_i + NO \rightarrow \ldots \rightarrow N_2 \qquad (10)$$

10.2 The Relative Importance of Thermal and Fuel NO_x

The relative importance of thermal and fuel NO_x has been established by burning the fuel of interest in a molecular nitrogen-free atmosphere (achieved by replacing the combustion air with a synthetic oxidant mixture containing oxygen (O_2), carbon dioxide (CO_2), and argon (AR)). The O_2 content was maintained at 21 percent by volume and the CO_2/Ar ratio was selected to match the theoretical flame temperature to that obtained with air. The use of synthetic oxidants for fuel NO definition has been verified by a variety of experimental studies.(48)

Figure 10-1 shows data obtained by England et al(49) for a low nitrogen distillate oil (No. 2) and a heavy residual oil (No. 6). Thermal NO_x formation was dominate in the case of the distillate oil because of the extremely low fuel nitrogen content; however, with the heavy oil, both mechanisms were of major importance.

Figure 10-2 is a composite plot of total and fuel NO_x emissions for a wide variety of liquid fuels tested at 5 percent excess O_2 and 405K preheat in a 21 kW bench-scale furnace obtained by England et al.(49) Results obtained with petroleum-derived fuels are presented in Figure 10-2a. Both total and fuel NO_x emissions increase with increasing fuel nitrogen content. Distillate oils doped with pyridine and thiophene (slashed symbols) correlate well with the actual heavy petroleum liquids under the conditions of these tests (fine oil

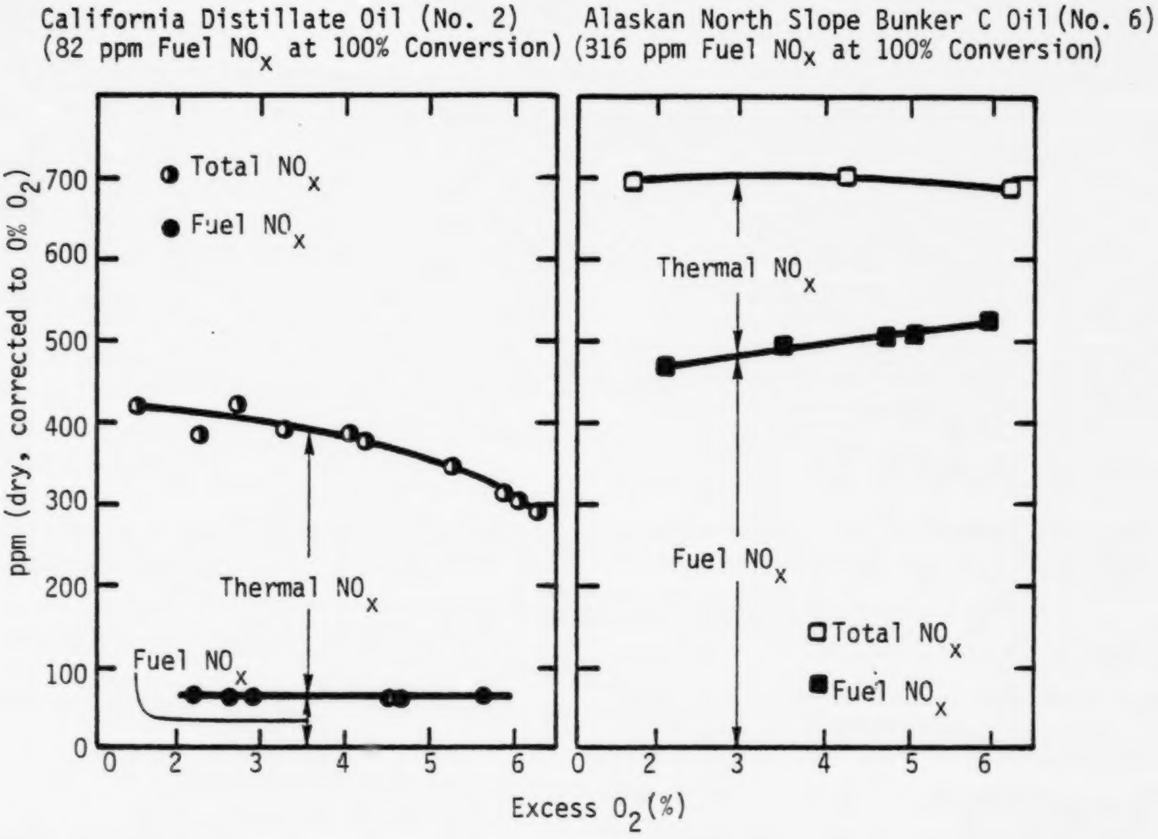
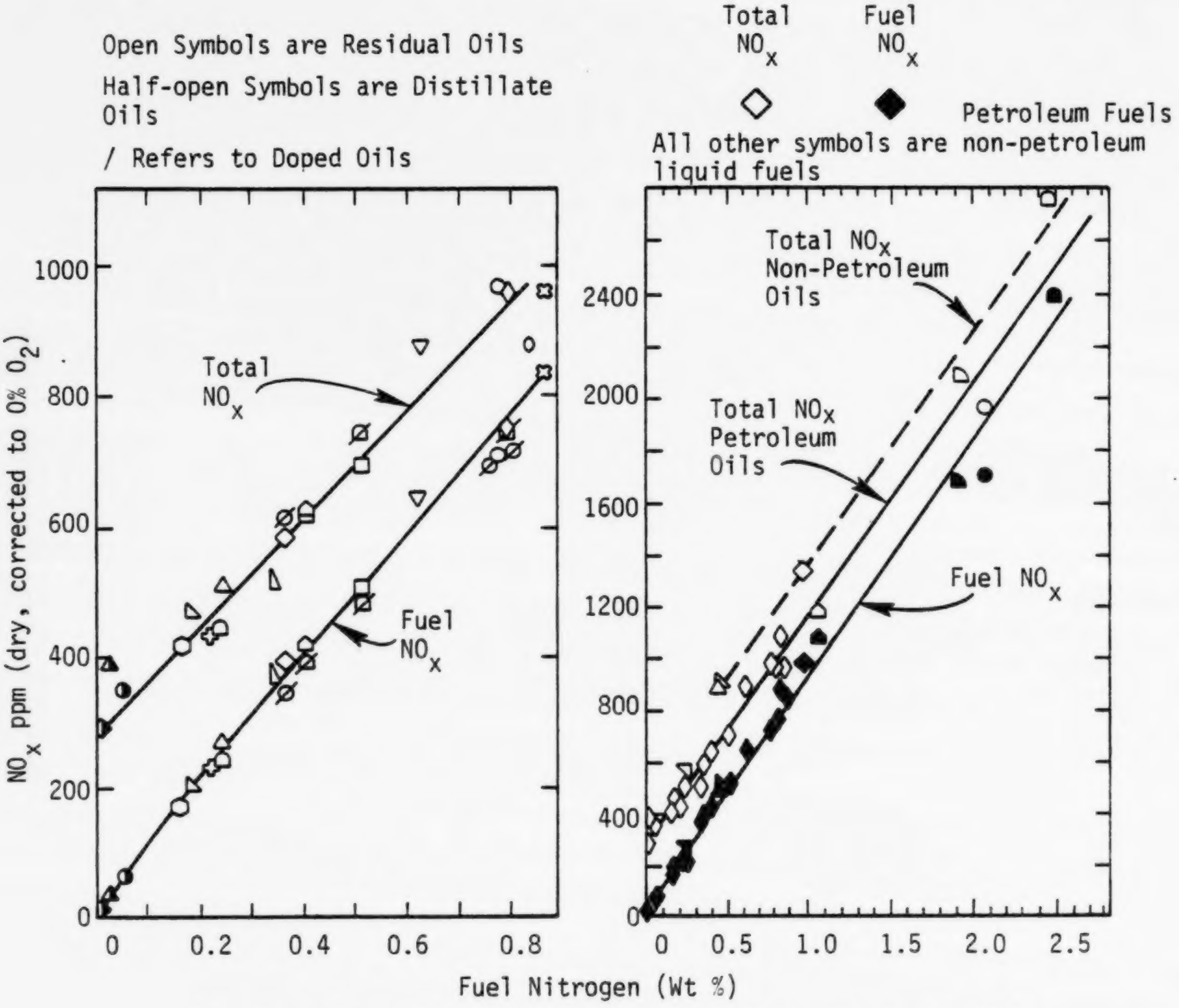


Figure 10-1. Total and Fuel NO$_x$ Emissions - Liquid Fuels.



(a) Petroleum Oils (b) All Liquid Fuels

Figure 10-2. Total and Fuel NO$_x$ Produced by Liquid Fuels (England et al[49]).

atomization, rapid mixing burner). The difference between total and fuel NO_x data is defined to be thermal NO_x and is approximately constant for all the heavy residual oils. However, thermal NO_x produced from distillate oils is consistently higher than that produced from residual oils.

A comparison of total and fuel NO_x produced with alternative and petroleum-derived fuels is presented in Figure 10-2b. In general, alternative fuels have higher nitrogen contents than petroleum-derived fuels, and, therefore, produce higher fuel NO_x emissions. With few exceptions, fuel NO_x emissions from both alternative and petroleum-derived fuels appear to correlate well on the basis of fuel nitrogen content. Total NO_x emissions from alternative fuels (dashed line) are higher than from pure petroleum-derived fuels, suggesting a greater production of thermal NO_x. This may be because the alternative fuels contain significantly larger fractions of light hydrocarbons which cause higher peak flame temperatures since more fuel is burned prior to vitiation of the combustion air with recirculated combustion products, or because the relatively high vaporization rates produce higher local combustion intensities.

10.3 Mechanisms of NO_x Control Via Combustion Modifications

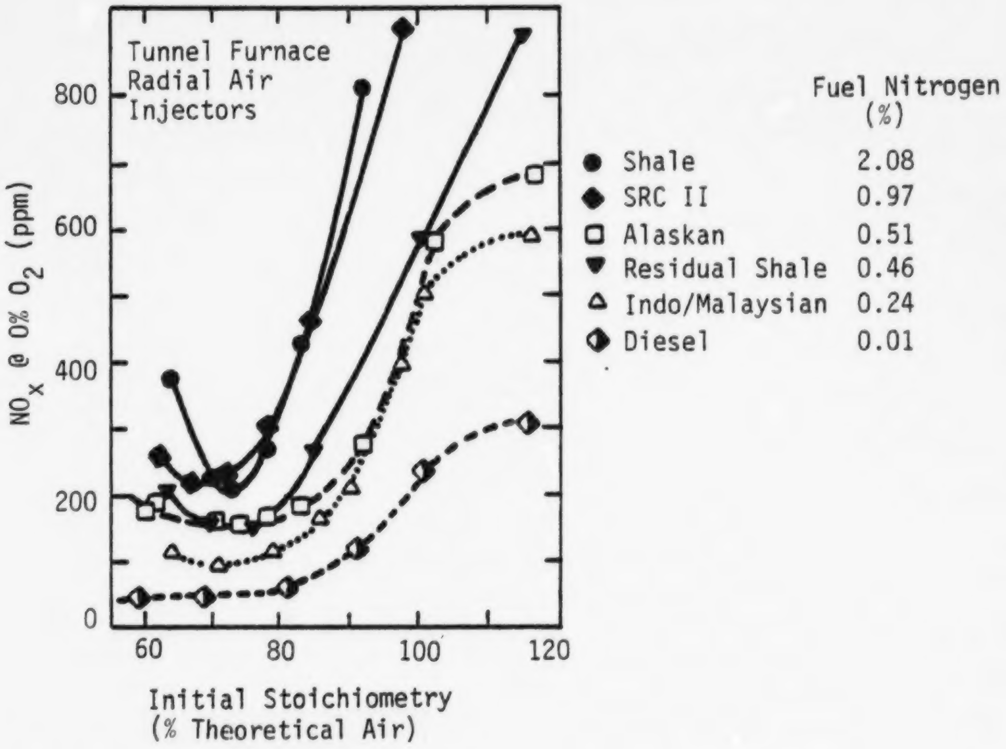
The previous sections discussed the fate of fuel-bound nitrogen during the combustion of liquid fuel, and the availability of oxygen was noted to be one of the major factors controlling fuel NO_x formation. The discussion on gas-phase chemistry described the mechanisms whereby nitrogen species are converted to NO_x or N_2, and the latter path predominates in fuel-rich systems. Consequently, modification of the combustion process to allow the oil to react initially under oxygen-deficient conditions (staged combustion) is the most effective method of controlling NO_x formation in combustors fired by fossil fuel. Staged combustion can be accomplished by several different techniques. The combustion process can be staged either by physically separating the fuel-rich zone from the burnout zone, or by using the burner to delay the mixing of the fuel with all the combustion air. This section focuses on the chemistry of staged combustion; the actual hardware details will be discussed later in this report.

10.3.1 Primary Zone Stoichiometry

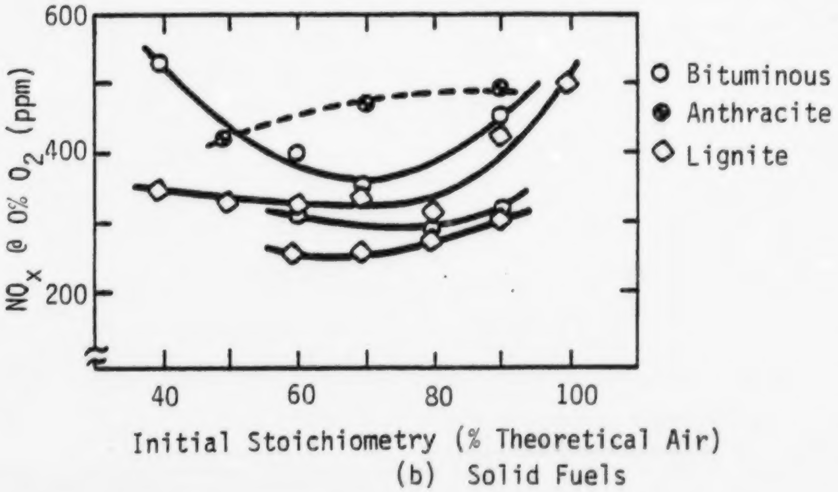
Figure 10-3 presents results from staged combustion experiments which were carried out in bench-scale furnaces operated with a fuel-rich primary zone (residence time about 1 sec) and with downstream air injection. The exhaust NO_x emissions are reported as a function of the initial (primary zone) stoichiometry (percent theoretical air). The overall excess air was maintained constant, and results are presented for several solid and liquid fuels. As the primary stoichiometry was decreased, the exhaust NO_x first decreased, reached a minimum, and then for nitrogen-containing liquid fuels, increased (Figure 10-3a). This minimum was much more pronounced with nitrogen-containing liquid fuels than with the solid fuels (shown for comparison) and for the non-nitrogen fuel (diesel oil) no minimum in exhaust NO_x was observed.

During fuel-rich combustion, the original fuel nitrogen species are ultimately converted to N_2, HCN, NO, or NH_3. The intent of staged combustion is to minimize the concentration of total fixed nitrogen (NH_3 + NO + HCN = TFN) prior to second-stage air addition. Figure 10-4 presents data obtained from detailed in-flame measurements for a distillate oil (Alaskan diesel), a typical residual oil (Alaskan Bunker C), and an alternate liquid fuel (SRC-II heavy distillate). These measurements were made on the center line of the furnace at a distance of 103 cm (approximately 630 msec) from the oil nozzle. Detailed radial measurements indicated that the concentration profile was essentially uniform at this location. These in-flame data are reported on a dry, as-measured basis and include nitrogen species (NO, NH_3 and HCN) as well as total hydrocarbons (THC). Second-stage air was added at 107 cm subsequent to each in-flame measurement in these experiments to bring overall theoretical air to 116 percent (3 percent O_2). Exhaust NO_x measurements $(NO_x)_E$, are also shown dry corrected to zero percent oxygen.

Decreasing the first-stage stoichiometry ratio reduced the NO concentration leaving the first stage. However, below a stoichiometry of approximately 80 percent T.A., significant amounts of THC and HCN were measured. Thus the minimum in exhaust NO_x concentrations (shown in Figure 10-3) is due to a competition between decreased first-stage NO and increased oxidizable nitrogen species such as HCN. It should be noted that the HCN concentration increased rapidly below 75 percent T.A. and that this increase was accompanied by an

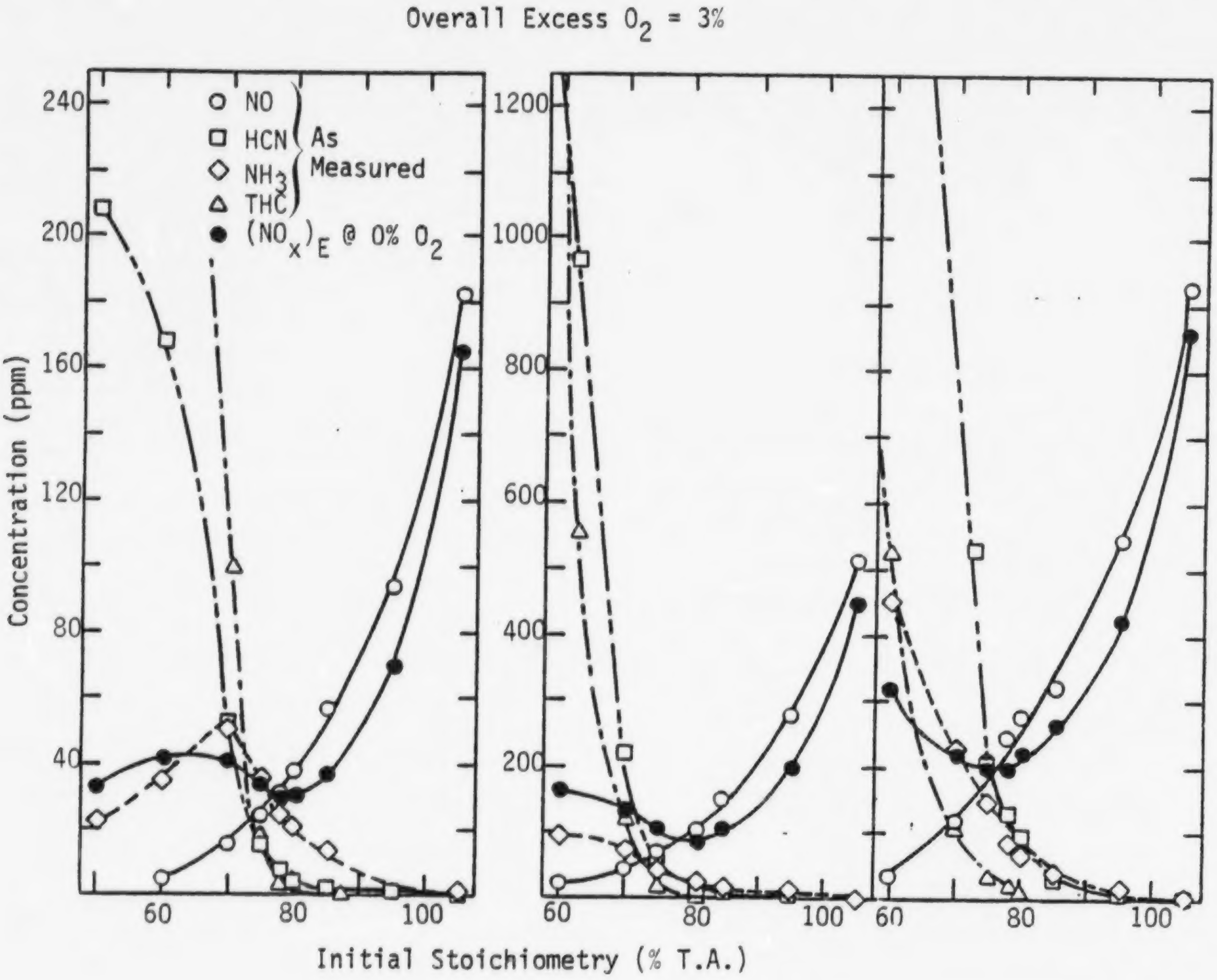


(a) Liquid Fuels



(b) Solid Fuels

Figure 10-3. Effects of Staging for Solid and Liquid Fuels (Heap[50]).



(a) Alaskan Diesel Oil (b) Alaskan Bunker Oil (c) SRC-II Heavy Distillate Oil

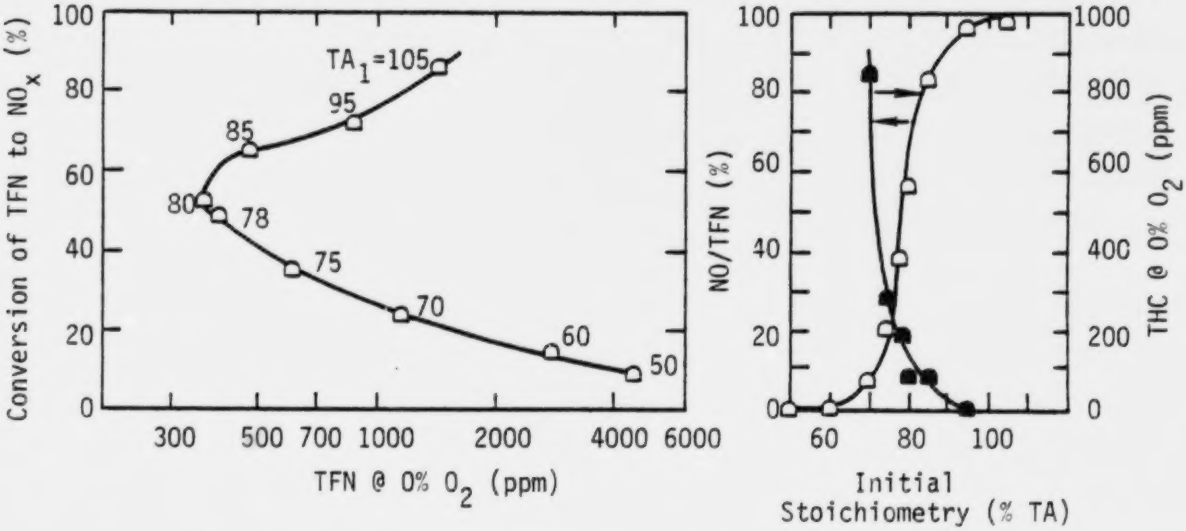
Figure 10-4. NO, HCN and NH_3 Production in the Fuel-Rich Primary Zone (England et al.[49]).

increase in hydrocarbon content of the partially oxidized combustion products. Data for the Alaskan diesel oil (Figure 10-4a) also show the presence of much smaller, but significant, concentrations of HCN and NH_3, although this fuel is essentially nitrogen-free. (Total conversion of the fuel nitrogen would produce 15 ppm TFN at 70 percent theoretical air.)

Exhaust NO_x emissions in a staged combustor result from conversion of the total fuel nitrogen (TFN) exiting the first-stage and any thermal NO_x formed during burnout. Thus, the optimization of first-stage stoichiometry is complex. Measurements within the fuel-rich first stage show that at the optimum stoichiometry NO, HCN, and NH_3 co-exist. TFN exists mainly as HCN and NH_3 for more fuel-rich conditions, and for conditions less fuel-rich than the optimum, NO is the primary fixed nitrogen species. Three factors influence the stoichiometry at which the minimum exhaust NO_x emissions occur: (1) the amount of TFN species present, (2) the relative conversion efficiencies of the TFN species to NO_x in the second-stage flame, and (3) the influence of other partial combustion products on TFN conversion.

Figure 10-5a shows the conversion of the first stage TFN to NO_x for a shale oil. As the first-stage stoichiometry is progressively reduced from 105 percent T.A., the TFN decreases (due to decreased first-stage NO formation) and the percent conversion also decreases (due to increased second-stage combustion). These two effects result in a rapid decrease in exhaust NO_x. Below 80 percent T.A., the TFN begins to rise dramatically because of HCN and NH_3 formation, but the conversion of TFN to NO_x continues to decrease. Thus, in general, there are two first-stage stoichiometries with identical TFN levels (e.g., 1000 ppm) but with vastly different conversion efficiencies (77 percent versus 26 percent) and hence, different exhaust emissions.

Figure 10-5b indicates that there are two major differences between the upper and lower percent conversion curves. First, below approximately 80 percent T.A. the major TFN species shifts from NO to HCN. Second, at first-stage stoichiometries less than 80 percent T.A., significant amounts of unburned hydrocarbons are carried into the second-stage flame. Folsom(51) has shown that in a simple diffusion flame the conversion of HCN and NH_3 to NO and the retention of NO are affected by the hydrocarbon content of the fuel. NH_3 conversion is low when hydrocarbons are absent, and hydrocarbons are necessary



(a) Conversion of TFN to NO_x versus TFN and Initial Stoichiometry

(b) NO and THC in First Stage versus Initial Stoichiometry.

Figure 10-5. Effects of First Stage Exit Composition on Second Stage NO_x Formation for Shale Oil (England et al[49]).

for the reduction of NO or HCN to N_2. Thus, staged combustion requires the minimization of TFN, as well as the production of an optimum TFN/fuel mixture for burnout.

10.3.2 Fuel Chemistry

Figure 10-6 summarizes the minimum emission levels observed with the various liquid fuels tested in the bench-scale furnace by England et al.(49) This figure illustrates that the minimum emission levels observed under staged combustion conditions do not correlate simply with total fuel nitrogen content which is in direct contrast to the results obtained under excess air conditions (see Figure 10-2). The data indicate that the doped distillate produces NO emissions which are significantly lower than a residual oil of the same nitrogen content. The CH_3/NH_4 emissions were also dramatically lower than the high nitrogen liquid fuels tested. The shale oil does not correlate with the petroleum-derived liquids. In summary, the data shown in Figure 10-6 suggests that the effectiveness of staged combustion cannot be simply assessed based on a knowledge of the fuel nitrogen content of a particular liquid fuel.

The variation in minimum NO_x levels with fuel composition is partially related to the evolution and subsequent oxidation of volatile nitrogen species. Figure 10-7 shows the partition of TFN among HCN, NH_3, and NO at the end of the fuel-rich first stage, as a function of the first-stage stoichiometry. Data for the six liquid fuels and 0.79 weight percent nitrogen (as (NH_3) in methane gas are presented. In each case, the missing fuel nitrogen has either been converted to N_2 or still exists in some partially oxidized fuel fragment. All of the liquid fuels exhibited the same general trends:

- Above approximately 80 percent theortical air, NO was the dominant TFN species.
- Below 75 percent theoretical air, the major TFN species was HCN and under very fuel-rich conditions, HCN usually accounted for more than 50 percent of the initial fuel nitrogen.
- First-stage NH_3 levels were always low, generally less than 20 percent of the fuel nitrogen. Axial profiles indicated that near 80 percent T.A., significant amounts of NH_3 were formed early in the rich zone, but decayed rapidly.

Alternative liquid fuels produced TFN distributions very similar to those

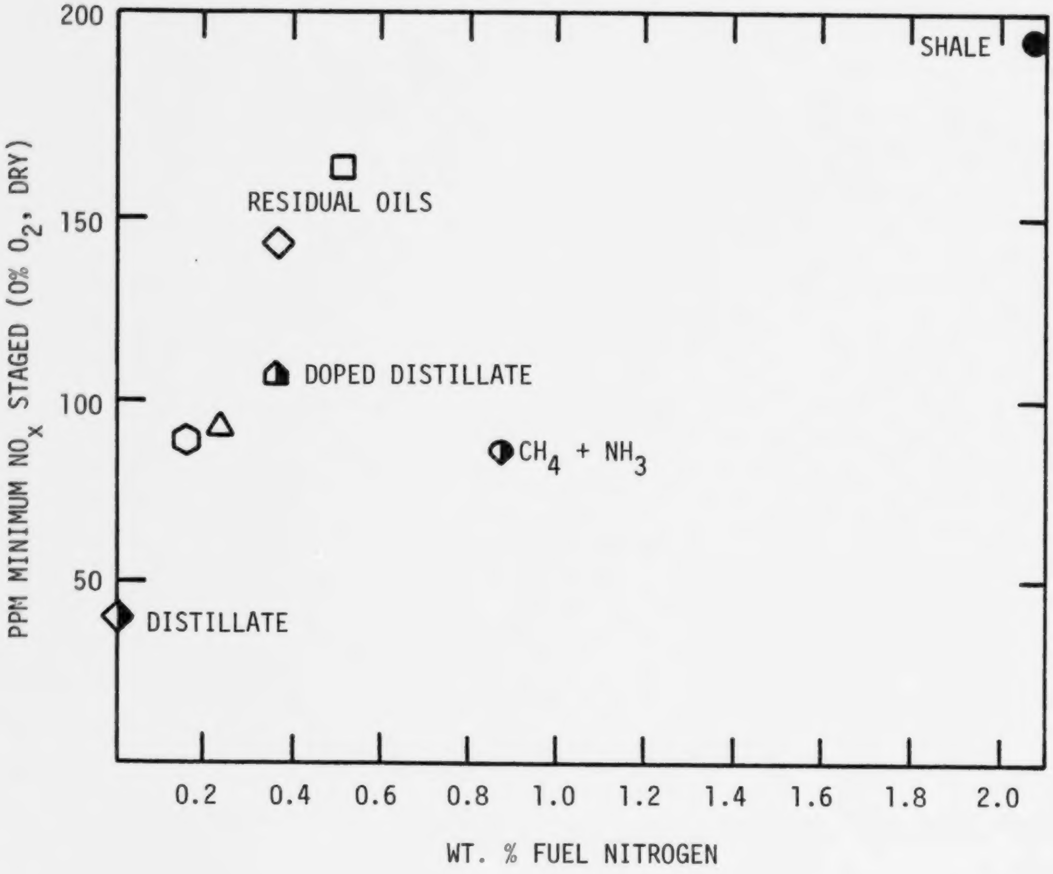
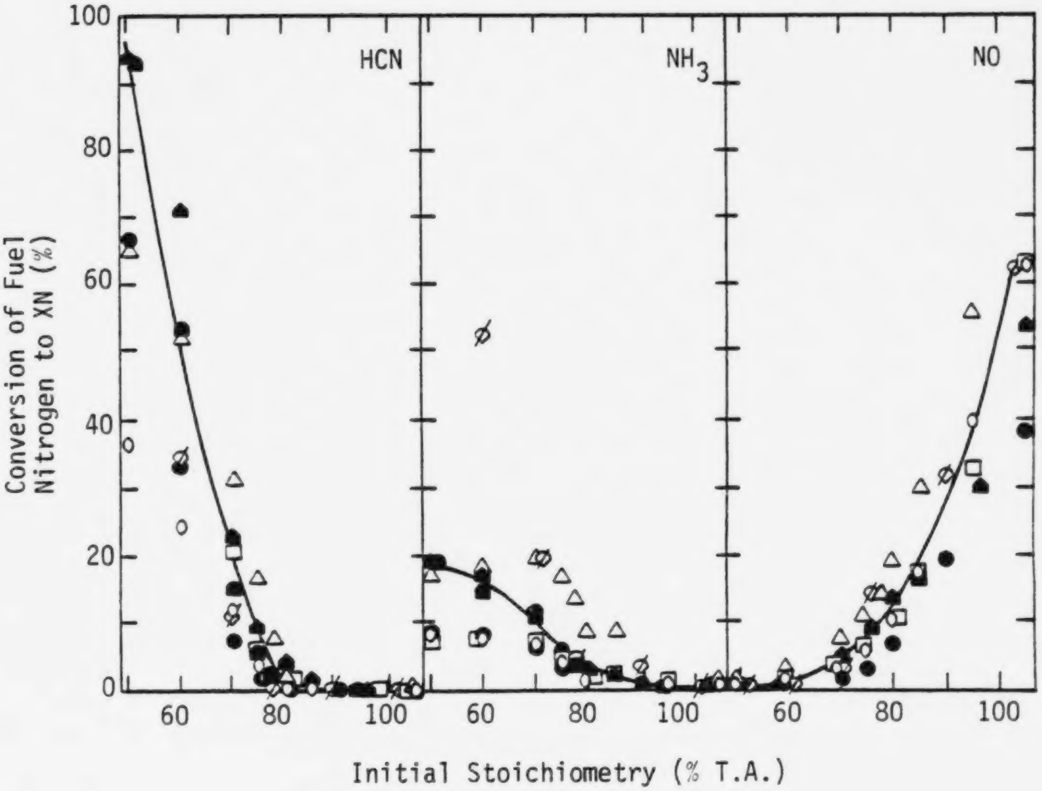


Figure 10-6. Influence of Fuel Nitrogen Content on Minimum NO_x Emissions Staged.



Fuels Include Six Liquids and
a NH_3/CH_4 mixture (slashed symbol).

Figure 10-7. The Conversion of Fuel Bound Nitrogen to Gaseous Nitrogen Species.

observed with petroleum-derived liquids.

10.3.3 Thermal Environment

The TFN concentrations shown in Figure 10-4 are in excess of equilibrium levels, and Sarofim(52) has suggested that increasing the temperature of the primary zone would prove beneficial. The results presented in Figures 10-8 and 10-9 obtained with a shale crude demonstrate the impact of first-stage heat loss on the fate of fuel nitrogen. The hot conditions refer to the furnace without primary zone cooling. To cool the furnace, cooling coils were added in the first stage. Figure 10-8 shows exhaust NO_x and primary zone gas temperature as a function of stoichiometric ratio for the two conditions. Minimum NO_x emissions were reduced for the hot case, and the optimum stoichiometry was shifted toward more fuel-rich conditions.

The axial profiles presented in Figure 10-9 provide an explanation for this shift in the minimum emission levels. Heat extraction in the first stage impacts the rate of decay of TFN. Under cold conditions both NO and HCN essentially freeze, whereas without heat extraction the initial rate of decay for all three species is much faster, leading to low TFN concentrations at the exit of the fuel-rich first stage. It should be noted that heat extraction also affects the rate of CO oxidation. Thus, in liquid fuel (and nitrogen-containing gas) flames, primary zone heat removal should be minimized because elevated temperatures accelerate the rate of TFN species decay toward low equilibrium levels.

10.4 NO Formation in Fuel Oil Flames

The two factors controlling NO_x formation in residual fuel oil flames are flame temperature and oxygen availability. Fuel and air are supplied separately, and all the fuel does not necessarily react at the overall stoichiometry of the input mixtures. The flame is confined in a combustion chamber, and the reactants are progressively diluted with recirculating combustion products and because of the turbulent nature of the flow, some fuel reacts initially under fuel-lean conditions. Other elements of fuel pass through high temperature combustion products before air is available to allow combustion to take place and therefore experience prolonged times in fuel zones. Additionally, the liquid fuel must be atomized (i.e., broken into small droplets). If the fuel is not atomized sufficiently, particulate and smoke emissions tend to

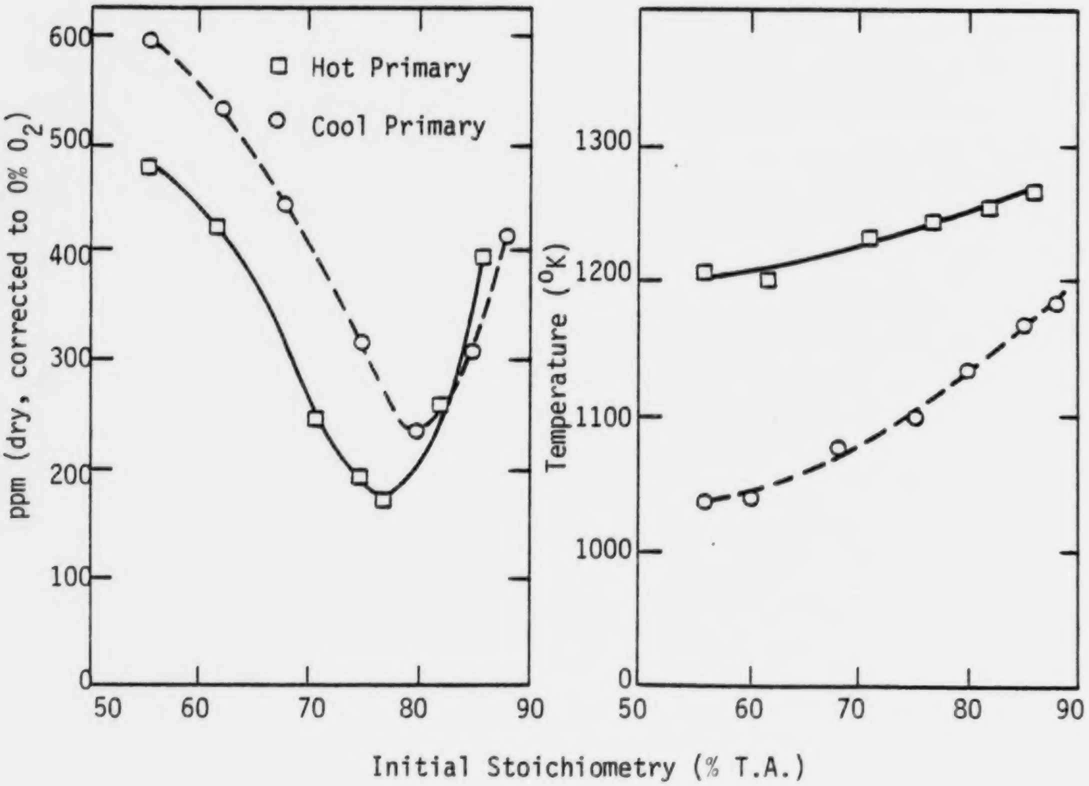
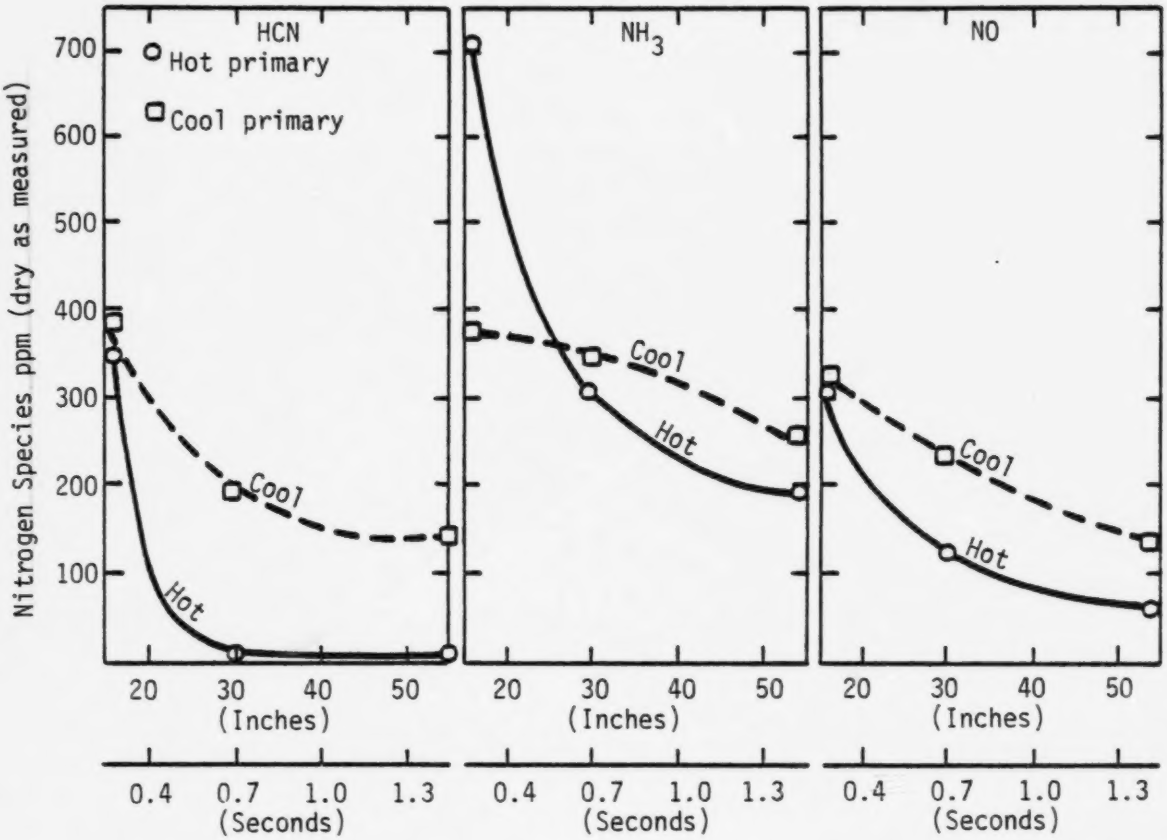


Figure 10-8. The Influence of Fuel Rich Primary Zone Temperature Profile on Exhaust NO_x (England et al[49]).



Position or Residence Time in Fuel Rich First Stage

Figure 10-9. Effect of First Stage Cooling Upon Nitrogen Species for a Liquid Fuel (Heap et al[50]).

be high because of inadequate fuel air mixing.

The conditions existing at the base of a residual fuel oil flame are shown conceptually in Figure 10-10 for two limiting conditions. These conditions are dependent upon the residence time of the liquid spray in an internal recirculation zone. This recirculation zone could be produced because of the divergence of flow caused by a swirling combustion air flow, or because of the use of a bluff body flame stabilizer. In one case (Figure 10-10a), the spray is assumed to have a relatively narrow angle and its trajectory provides a relatively long residence time within the internal recirculation zone. This recirculation zone will consist of hot combustion products deficient in oxygen. The liquid fuel will vaporize, mix with the inert products, and combustion occurs at the boundary of the recirculation zone when air is available. In contrast to this condition, if the spray angle were wide (Figure 10-10b) the spray has a very short residence time within the recirculation zone. Small drops and vapor are entrained with the combustion air prior to entraining high temperature products which will provide an ignition source. The major difference between the two limiting cases is that in (a) the fuel vaporizes in the absence of oxygen, while in (b) the spray is first mixed with air before high temperature combustion products are entrained and the droplets vaporize. If the angle in (b) is too wide, then the flame may not be stable. However, if a stable flame is achieved these are normally the conditions which will maximize NO production since oxygen is available when the fuel nitrogen compounds are released from the liquid droplets. Alternatively, condition (a) is one which will tend to minimize fuel NO production since the droplets vaporize in the absence of oxygen and heat release takes place in a diffusion flame region, thus providing optimum conditions for the conversion of XN to N_2.

The temperature of the flame zone and, therefore, the rate production of thermal NO will be controlled by the stoichiometry and enthalpy of the reactants within the flame zone. Thus, if cooled external recirculation can be entrained prior to major heat release, the bulk flame temperature will be reduced and thermal NO production inhibited.

The above explanation is an oversimplification of a complex process. In real flames the fuel/air contacting process will be controlled by such factors as the axial and tangential momentum of the combustion air, the momentum of the

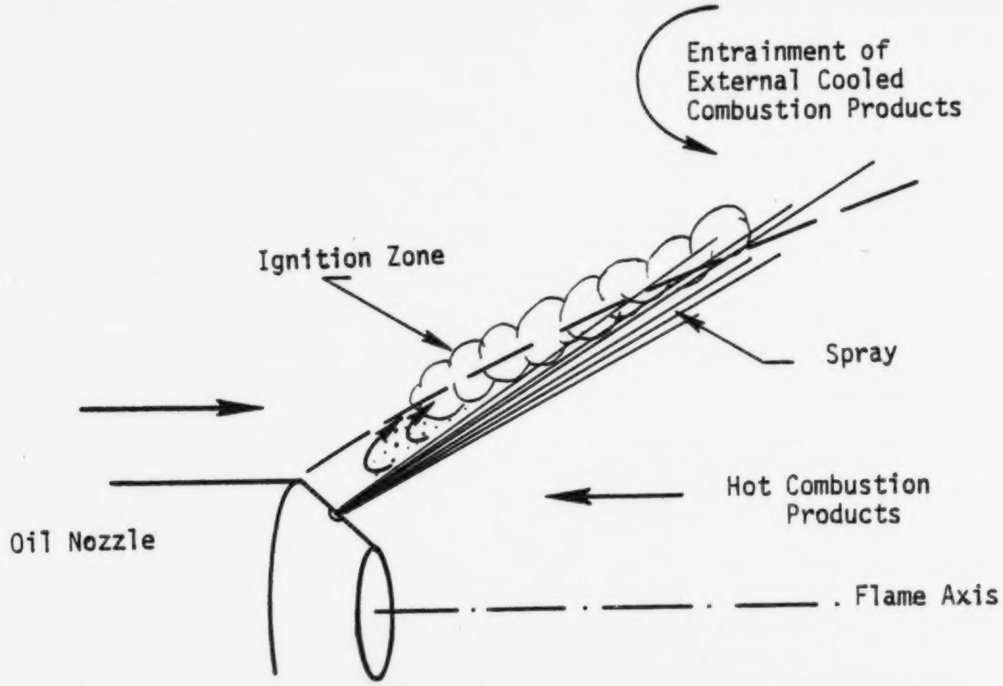

Entrainment of
External Cooled
Combustion Products
Ignition Zone
Spray
Hot Combustion
Products
Oil Nozzle
Flame Axis

Figure 10-10a. Spray Trajectory Passes Through the Internal Recirculation Zone.

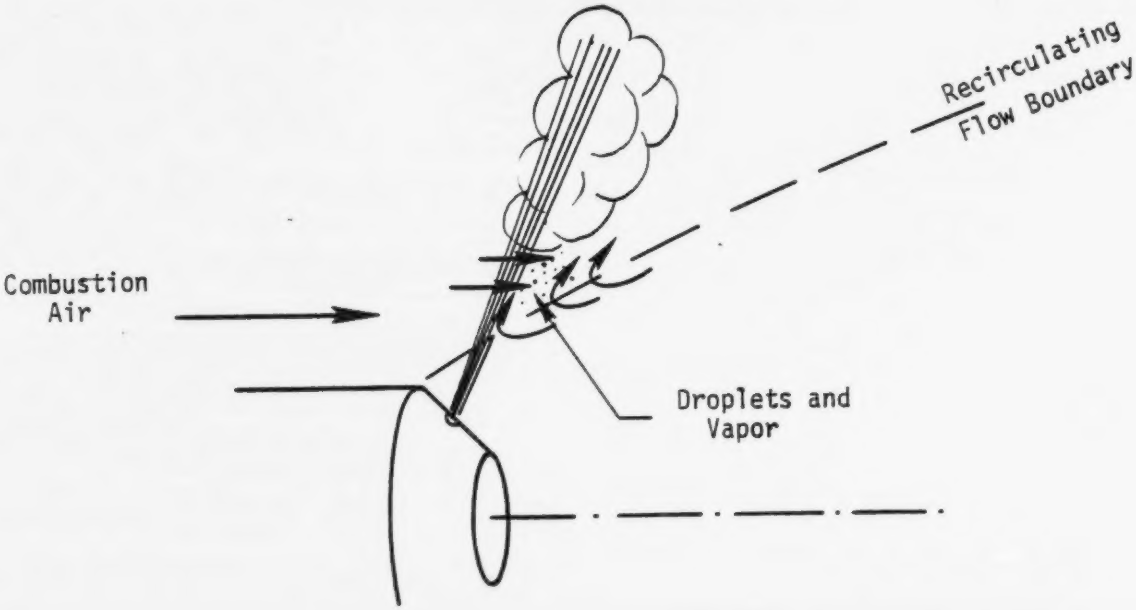

Recirculating
Flow Boundary
Combustion
Air
Droplets and
Vapor

Figure 10-10b. Spray Rapidly Mixes With Air.

Figure 10-10. Concept of Fuel Air Contacting at the Base of a Residual Oil Flame.

fuel jet and its trajectory, the drop-size distribution of the liquid fuel spray, and the strength and size of the internal recirculation zone. Knowledge of those parameters controlling fuel/air contacting allow the development of combustion systems which will provide those conditions under which both thermal and fuel NO_x formation are limited. The burner design should also provide a flame shape which is compatible with the combustion chamber. Thus burners suitable for firetube boilers may not be acceptable in water boilers.

The preceding discussion clearly defines the important role that the fuel oil atomizer plays in the combustion of liquid fuels and the development of flame characteristics. A review of the available literature concerning the influence of atomization parameters on pollutant emissions from oil-fired equipment produces conflicting evidence concerning the importance of these parameters. Cato et al(3) concluded that a well-maintained oil burner operating at near its design point will produce about the same level of nitrogen oxides and particulate emissions regardless of the atomization method used, and that boiler efficiency was unaffected to any significant degree by the type of atomization employed. For a given oil burner the oil atomization method that produced the lowest nitrogen oxide emissions usually produced the highest particulate emissions. Figure 10-11 summarizes a series of field tests on one boiler with two No. 6 oils (nitrogen contents of 0.44 and 0.14) and a No. 2 oil. It can be seen that with the nitrogen-containing fuel steam atomization tended to give higher NO_x emissions, however, this trend was reversed with the No. 2 fuel oil. Figure 10-12 compares NO_x emissions from a laboratory combustor with one nozzle and one fuel as a function of atomization fluid and pressure. These results indicate that at a given atomization pressure, steam atomization gives lower NO_x emissions than air atomization. However, this conclusion is not independent of pressure.

EPA-supported investigations at the IFRF(22) showed a dramatic effect of atomizer design on NO_x emissions. This investigation was carried out in a refractory tunnel furnace with a double concentric swirl burner and compared NO_x emissions from various atomizer designs and characteristics (steam assist, mechanical spray angle) for a given firing rate and fuel oil. The results presented in Figure 10-13 show that NO_x emissions can be reduced from approximately 230 ppm to around 100 ppm by fitting a distributor tip over a normal Y-jet atomizer in order to divide the flame. The distributor tip divides the

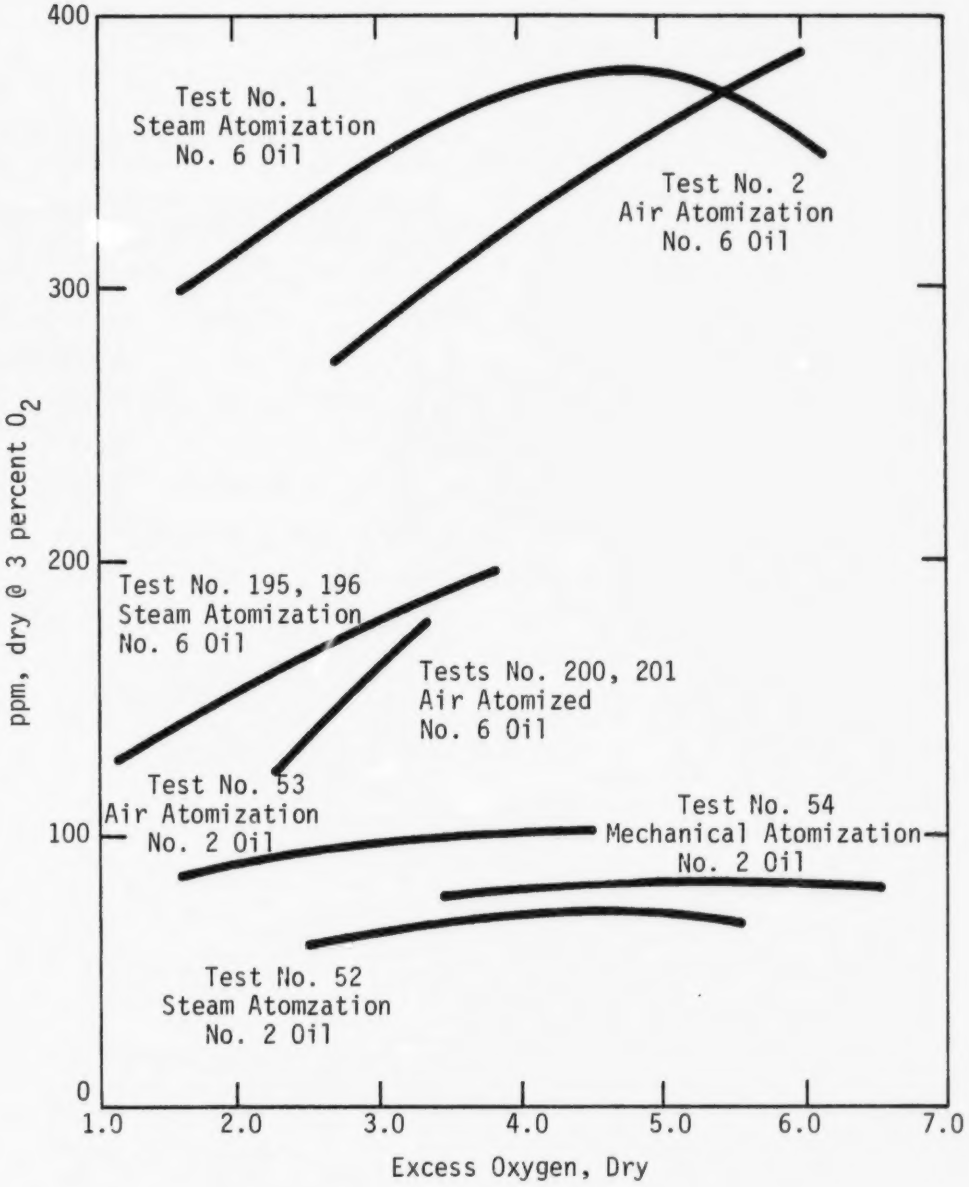


Figure 10-11. Effect of Oil Atomization Method and Excess Oxygen Level on the Total Nitrogen Oxides Emissions (after Cato et al[3]).

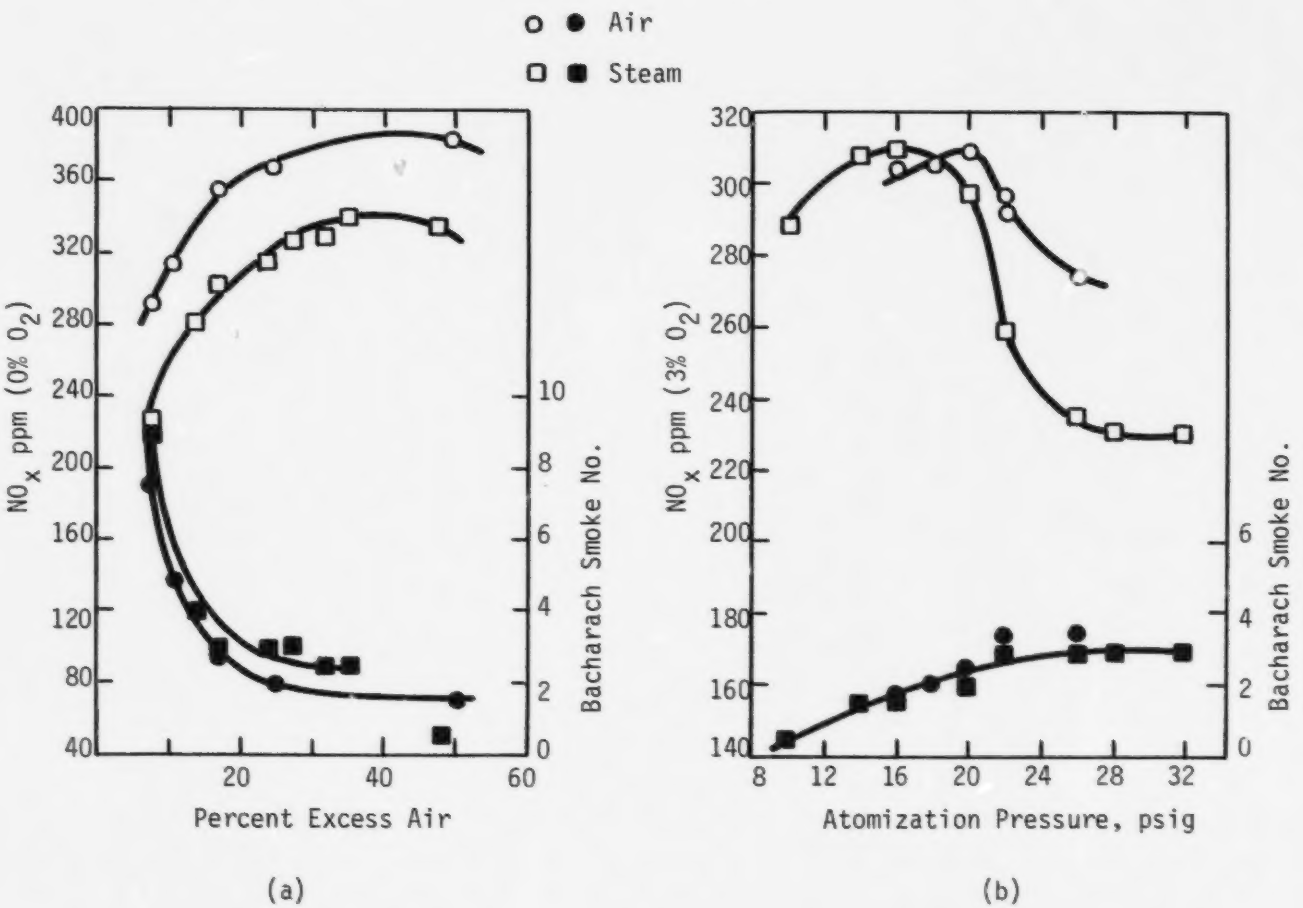


Figure 10-12. (a) NO$_x$ and Smoke Emissions as a Function of Excess Air Level and Atomization Medium and (b) The Influence of Atomization Fluid and Pressure Pollutant Emissions from Oil Flames (17 percent Excess Air, 3.4 x 10^6 Btu/hr).

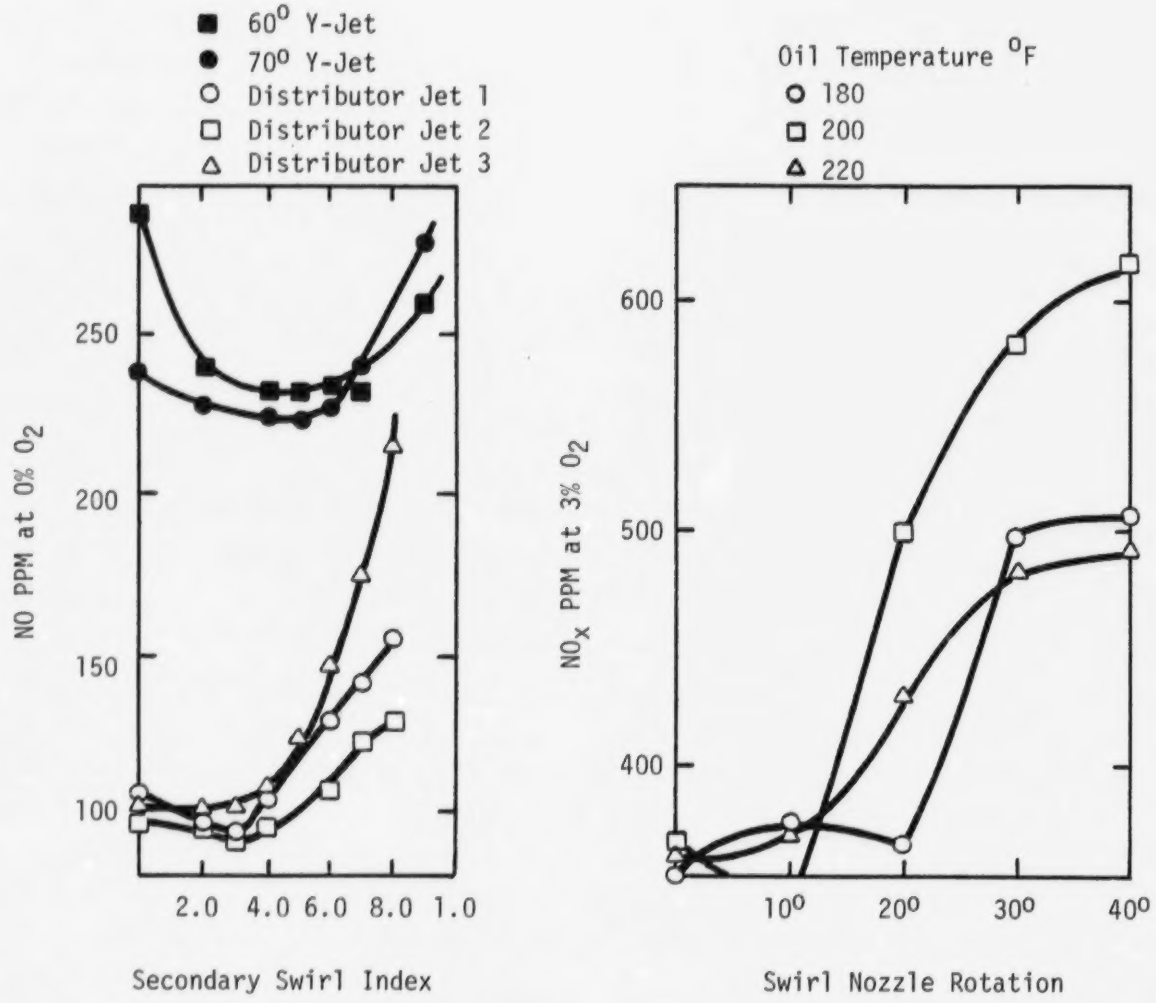


Figure 10-13. (a) Emission Characteristics of Fuel Oil Flames (5 percent Excess Air, 300°C Preheat, Primary Swirl Vanes 30°) and (b) The Effect of Fuel Oil Temperature on NO_x Emissions.

spray into a series of fuel jets which form individual flames contrasting with the single brush produced without the distributor tip. It can be argued that NO_x emissions are reduced because the individual flames will provide fuel-rich conditions within the flame jet, thus minimizing fuel nitrogen oxidation. These investigations were carried out under relatively high wall temperature conditions atypical of package boilers (although the volumetric heat release rate was low) and it could be argued that this type of atomizer would tend to produce smoke if used in a cold wall combustion chamber since local stoichiometry also affects smoke and particulate formation.

The preceding discussion was not intended to provide an exhaustive review of information on the impact of atomizer properties on NO_x and particulate formation in fuel oil flames. However, it does serve to illustrate the following points:

- For a given burner the atomizer design can have a strong influence on both NO_x and particulate emissions.
- The influence of atomization parameters cannot easily be separated from the effect of the flow field on the fuel oil spray.
- There is a lack of information on the influence of atomizer parameters under staged combustion conditions.

10.5 Low NO_x Burner Design Options For Residual Fuel Oils

Some form of staged combustion is necessary to reduce significantly NO_x emissions from residual fuel oil fired combustors. As explained earlier in this section, NO emitted from this type of equipment is formed from both molecular nitrogen supplied with the combustion air and fuel bound nitrogen. The fraction of the total emission attributable to the oxidation of fuel bound nitrogen is dependent upon the total fuel bound nitrogen content. However, for most residual fuels fuel NO is at least fifty percent of the total emissions. Thermal NO formation can be reduced by decreasing flame temperature, but fuel NO formation is relatively insensitive to temperature. Consequently, flue gas recirculation is not an effective NO_x control technique for residual fuel oils. Fuel bound nitrogen conversion to fuel NO can be minimized by limiting oxygen availability by staging the combustion process. Therefore, an effective low NO_x burner design for residual fuels must involve some variant of staged heat release.

Two basic staged combustor designs can be envisioned, namely:

1. Physically staged systems. The fuel rich zone is separated physically from the staged combustion air. The low NO_x burner suitable for firetube boilers developed under contract 68-02-1500 had a refractory prechamber and can be classified as a physically staged system. Firetube boiler designs can readily accommodate this type of design. Physically staged systems have the further advantage that the conditions in the prechamber can be optimized to minimize TFN concentrations. However, the temperatures and residence times to achieve very low concentrations may require specialized designs which can no longer be considered as retrofit designs.
2. Aerodynamically staged systems. Burner design is optimized to delay the mixing of the fuel and air, thus creating a fuel rich zone. This approach has the advantage that it can be retrofitted to a wide range of watertube boilers without considerable expense.

The following section of this report describe a series of tests carried out with an aerodynamically staged system (a distributed mixing burner) to ascertain the level of NO_x control that could be expected with this approach. A decision was made to test this concept in a simulator and not in the field because:

1. Field test conditions are not conducive to an examination of a large number of parameters.
2. The tests in the simulator provide data which can be related to field practice.
3. Testing in the field is expensive.

11.0 AN EVALUATION OF AN OIL FIRED DISTRIBUTED MIXING BURNER

This section of the report describes a series of tests to determine whether a distributed mixing burner could be fired on oil and be optimized to minimize both NO_x and particulate emissions.

11.1 Test Burner

The conditions necessary for the minimization of NO_x emissions in coal fired boilers are similar to those required by low NO_x oil burners. The nitrogen compounds must be driven from the coal into the gas phase to react in the absence of oxygen and produce molecular nitrogen, just as the fuel nitrogen compounds in liquid fuels must react under oxygen deficient conditions. A circular burner has been developed under EPA sponsorship which burns pulverized coal in such a manner that NO_x emissions are minimized while maintaining flame characteristics and fuel burnout consistent with process requirements. This burner is called a distributed mixing burner (DMB) because the combustion air is distributed throughout the heat release zone which produces an initial fuel rich zone. The DMB concept as applied to coal is shown in Figure 11-1. The design features which are used to provide controlled distributed fuel/air mixing are:

- The fuel injector design. The coal-plus-primary air is injected into the furnace either from a circular injector, an annulus, or from a divergent cone. The velocity is completely axial or injected with a swirling component provided either by swirl vanes in the primary jet or by introducing some tangential motion downstream of the fuel injector. These burner parameters provide highly uniformly mixed fuel with primary air to prevent the production of excessively-rich or excessively-lean zones.

- Swirl component of the secondary air. The swirl component in the secondary airstream provides for the correct size and strength of the internal recirculation zone.

- Velocity ratio between the secondary and primary streams controls the rate of mixing in the initial fuel-rich stream.

- Burner exit geometry. The rate of divergence of the burner exit controls the properties of the internal recirculation zone such as

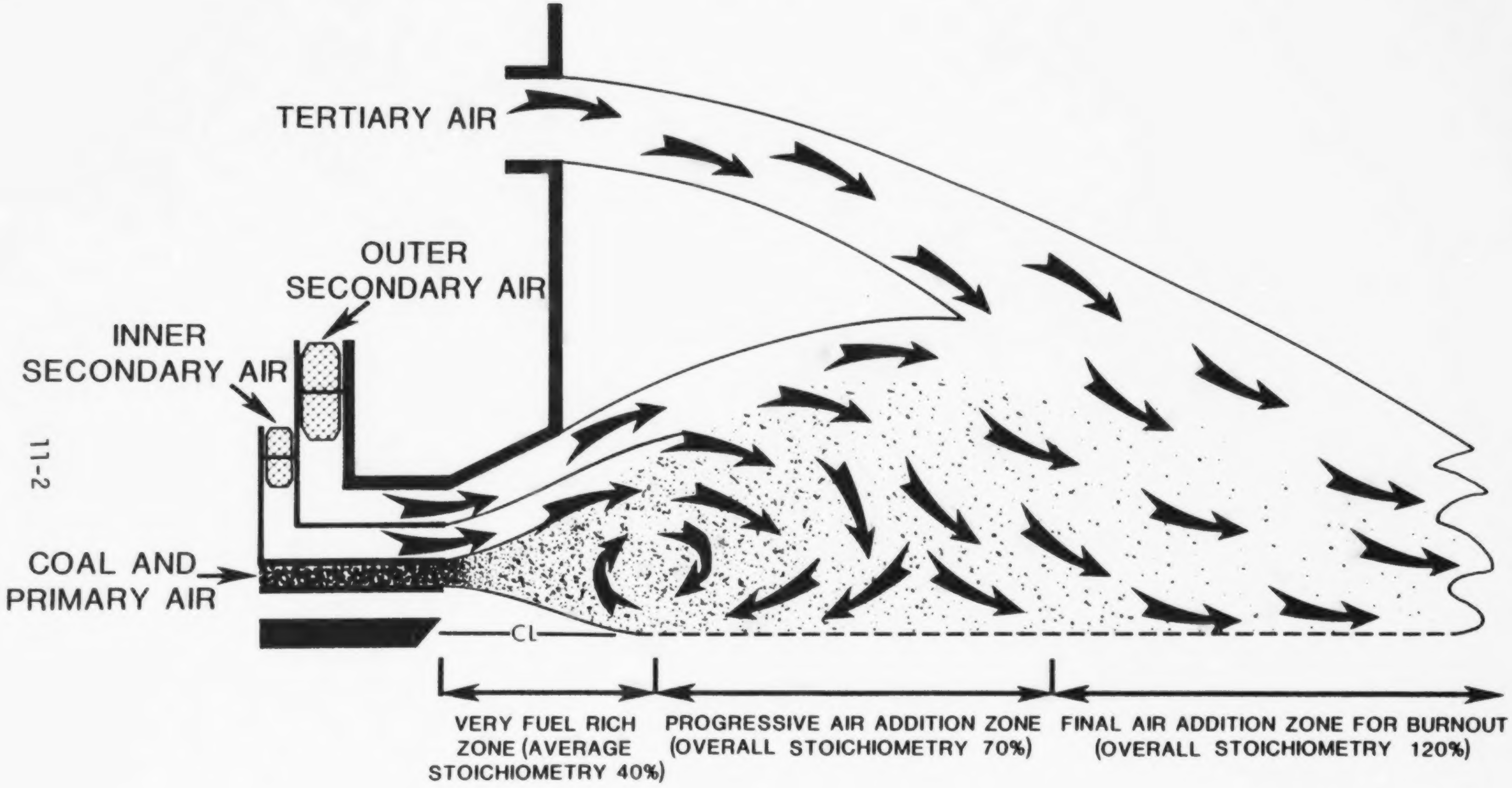

TERTIARY AIR
OUTER SECONDARY AIR
INNER SECONDARY AIR
COAL AND PRIMARY AIR
CL
VERY FUEL RICH ZONE (AVERAGE STOICHIOMETRY 40%)
PROGRESSIVE AIR ADDITION ZONE (OVERALL STOICHIOMETRY 70%)
FINAL AIR ADDITION ZONE FOR BURNOUT (OVERALL STOICHIOMETRY 120%)

Figure 11-1. Distributed Mixing Burner Concept for Pulverzied Coal.

size, strength and location.

- Location of the outboard staged air jets and momentum. These dictate the residence time in the rich section as well as the time available for burnout.

An oil ignitor is placed along the burner axis for use with coal. In this series of tests the ignitor gun was replaced with an appropriate size nozzle.

The specifications of the distributed mixing burner used in these tests are given in Table 11-1 and a sketch of the burner is shown in Figure 11-2. The burner design incorporates the following features:

- Three independently controlled airstreams to supply combustion air to:
  - the primary channel;
  - the secondary channel;
  - outboard staged air ports.
- A retractable oil gun situated on the burner axis suitable for use with air or steam atomization.
- Interchangeable annular swirl vanes to provide swirl to the primary flow.
- Adjustable annular swirl vanes in the secondary stream.
- Interchangeable refractory exits. Two exits were used with length to throat diameter ratios of 1 and 2.

Two different oil nozzle designs were used in this evaluation of the distributed mixing burner for oil firing, namely:

- Monarch

  The Monarch internal mixing atomizing nozzle is shown in Figure 11-3. The fuel oil and the atomizing fluid first mix in the swirl chamber before the spray exits around the pentle. The nozzle is manufactured in flow capacities ranging from 30-250 gph and is rated for No. 2 fuel oil but will atomize heavier oils if they are preheated to reduce the viscosity to 100 ssu. Spray angles from 50-100^0 are available and Monarch considers 80^0 the standard angle.

Table 11-1. Distributed Mixing Oil Burner Specifications

Nominal Firing Rate	10×10^6 Btu/hr
Primary Channel	
Outer Diameter	5 in.
Inner Diameter (Fuel Gun)	2 in.
Swirl Vanes	Variable
Area	15.5 in.^2
Secondary Channel	
Outer Diameter	9.5 in.
Swirl Vanes	Variable
Area	51.25 in.^2
Outboard Ports	
Location	22 in. for burner axis
Number	16/8
Injection Angle	Parallel
Diameter	4.5 in.

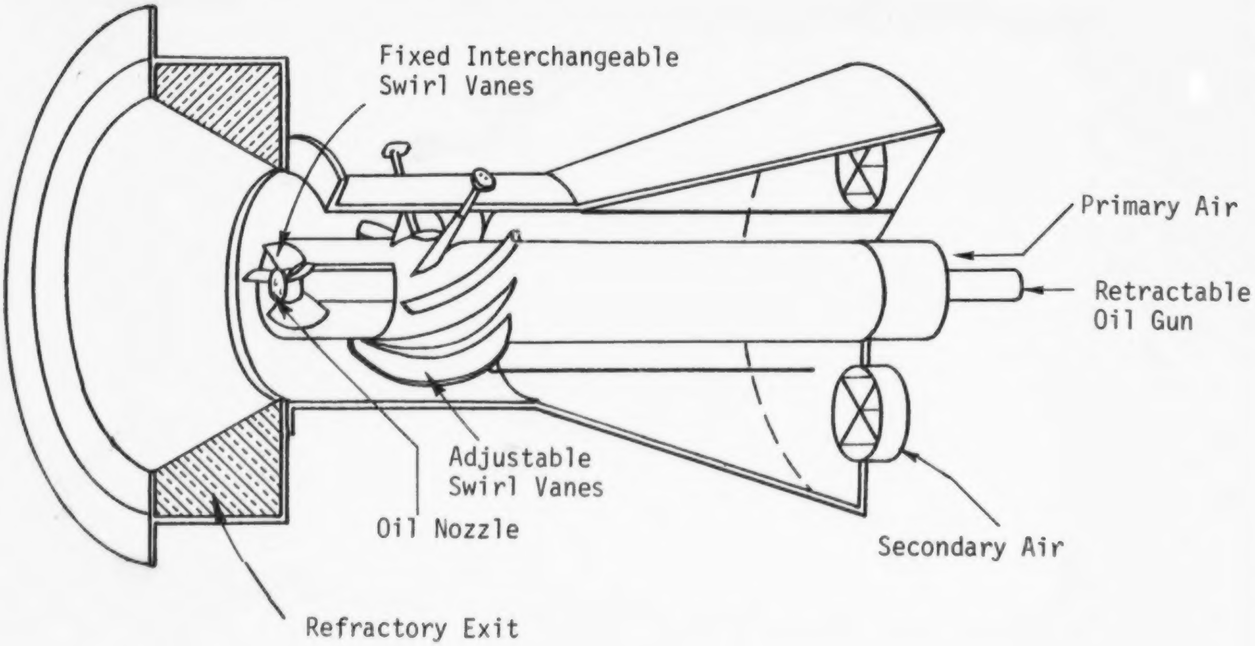

Fixed Interchangeable
Swirl Vanes
Primary Air
Retractable
Oil Gun
Adjustable
Swirl Vanes
Oil Nozzle
Secondary Air
Refractory Exit

Figure 11-2. Sketch of Distributed Mixing Oil Burner.
(Outboard Air Ports Not Shown)

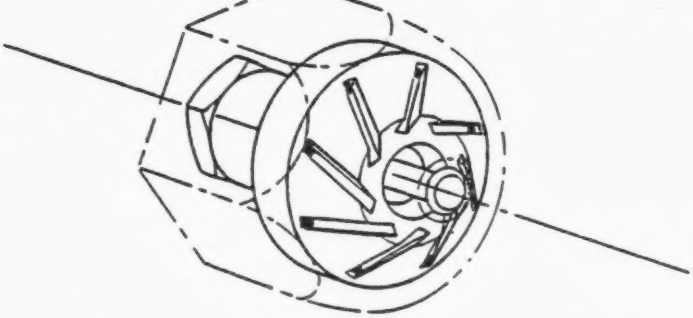


Figure 11-3. Monarch Nozzle.

- Todd

  The Todd-CEA twin-fluid atomized nozzle (Figure 11-4) operates with steam atomization only. Steam/fuel contacting is delayed so a preliminary mechanical atomization process can take place within a sprayer plate assembly. The fuel is injected into the mixing chamber at a right angle to the steam flow; the resulting steam/fuel mixture then leaves the mixing chamber through six holes which determine the spray angle.

11.2 Test Furnace

Development tests in the field are quite often impractical and expensive. The purpose of this investigation was to screen several design options which require frequent hardware changes. These type of tests would be difficult to arrange in the field because of the frequent interruption of steam supply. This problem can be overcome by conducting the tests in an experimental system which simulates the combustion chamber of a boiler. In addition, the use of a simulator allows a wider range of tests to be carried out for the same funds. However, for tests in the simulator to be meaningful, they should be carried out at a sufficiently large scale and in an environment typical of boilers in order that the information generated in the tests can be translated to field practices.

The tests which were carried out to establish the performance on an oil fired distributed mixng burner were carried out in a small watertube boiler simulator (SWS). The SWS was designed to simulate the thermal environment of a water tube boiler and also provide a similar geometric configuration. The trapezoidal cross-section is similar to D type boiler designs and the floor and firing wall were constructed of refractory. The roof and the remaining walls were constructed of sheet steel and were cooled by spraying water on them. The front wall was movable to vary the firing depth which in these tests was maintained at 11 ft. during these tests. The combustion products exited the combustion chamber from the side wall duplicating the entry to a convective section in a typical boiler. Combustion air could be preheated by a heat exchanger which was situated in the exhaust duct and a bypass system allowed the preheat temperature to be varied.

The simulator was fully instrumented to allow measurement of all input and output variables. The airflow to the various burner passages was measured

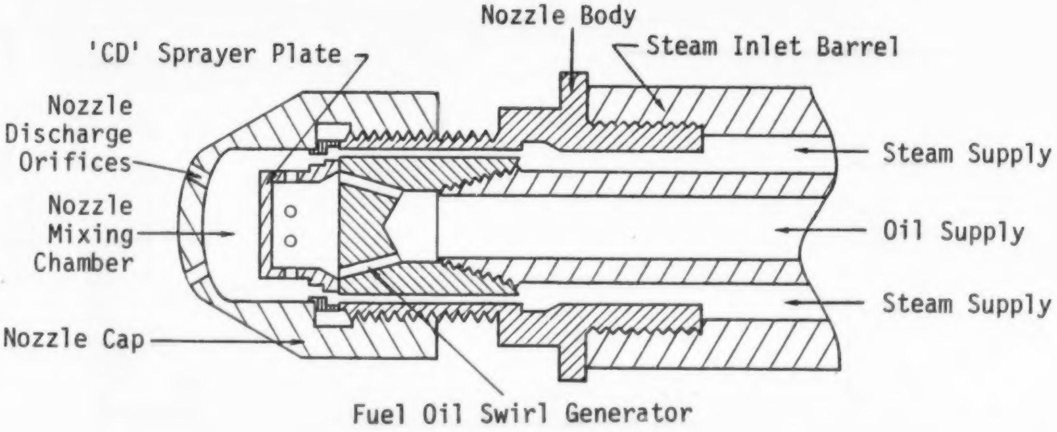

Nozzle Body
'CD' Sprayer Plate
Steam Inlet Barrel
Nozzle Discharge Orifices
Steam Supply
Nozzle Mixing Chamber
Oil Supply
Steam Supply
Nozzle Cap
Fuel Oil Swirl Generator

Figure 11-4. Todd-CEA Type 'CD' Atomizer.

by venturis and the preheat temperature was monitored by thermocouples. The oil flow rate and its pressure and temperature at the fuel nozzle were monitored continuously. The composition of the exhaust gases were monitored for O_2, CO_2, CO and NO/NO_x by commercial gas analyzers. The sample was withdrawn from the exhaust duct before the heat exchanger by a water cooled stainless steel probe and filtered and dried before being transferred to the analyzers by a Teflon line. The total sample rate was maintained at several times in excess of the instrument requirements to minimize response time. Excess sample flow was by-passed to a vent.

11.3 Results

The data obtained with the oil fired DMB are tabulated in Appendix B which lists all operating conditions and the results are discussed below in two sections corresponding to the two different nozzles. All tests were carried out with steam as the atomizing fluid and primary and secondary swirl vanes set at 45 and 60^o respectively.

11.3.1 Monarch Nozzle

Unstaged Performance

Figures 11-5, 11-6 and 11-7 illustrate the performance of the distributed mixing burner when operating in the unstaged mode with a relatively high nitrogen oil (0.77 percent). Unstaged emissions are approximately 500 ppm which is typical of projected emissions for field operating boilers with an oil of this nitrogen content(3). Emissions are highest at the highest load (see Figure 11-5), but they are not strongly affected by load below 9×10^6 Btu/hr. Increasing the secondary air preheat level to 180^oC did not impact emissions at high load (see Figure 11-6). In addition, emissions were relatively insensitive to the primary air flow rate (see Figure 11-7). Primary air is that air which is supplied in the channel around the oil gun; when operating as a coal burner the coal and transport air are supplied through this channel.

Staged Performance

The results presented in Figures 11-8, 11-9, 11-10, and 11-11 illustrate the performance of the oil fired distributed mixing burner operating with air flow through the outboard staged air ports. Thus the burner zone operates fuel rich providing the opportunity to prevent fuel nitrogen oxidation and thereby reducing NO_x emissions. The data presented in Figures 11-8 and 11-9

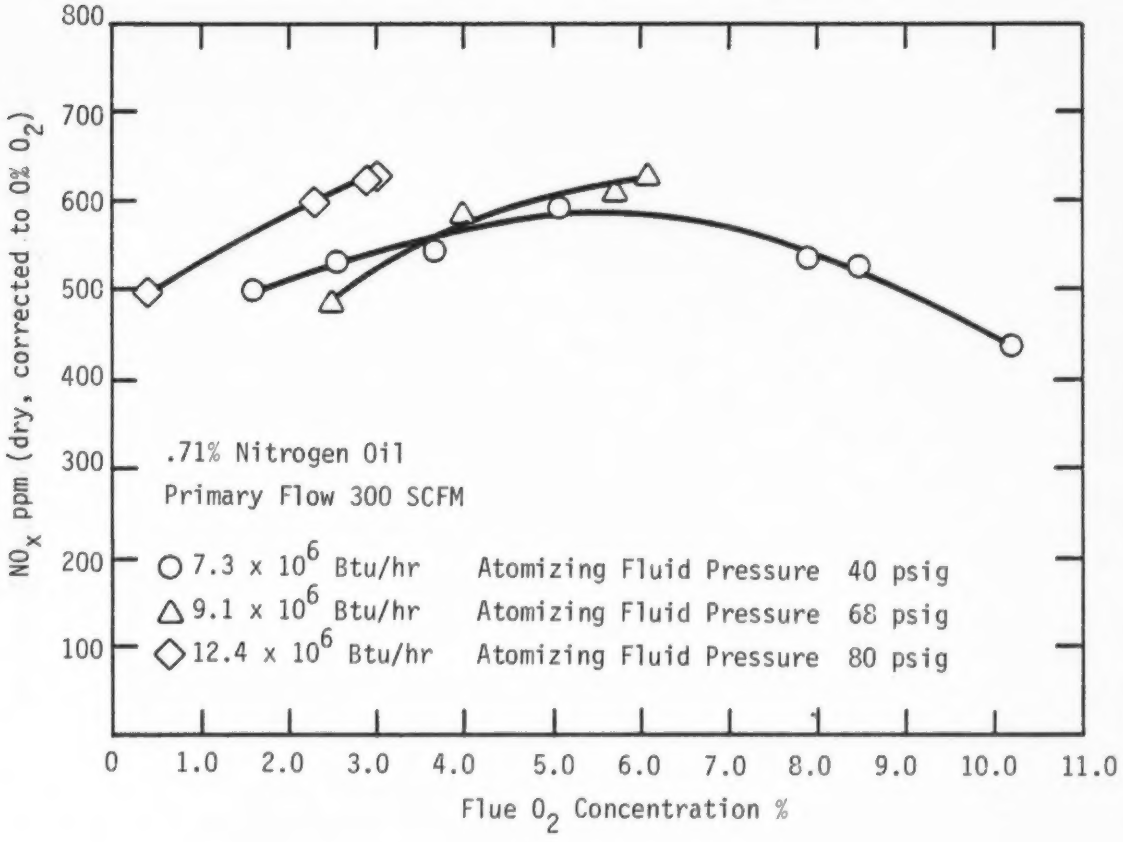


Figure 11-5. The Effect of Excess Air on Unstaged NO_x Emissions.

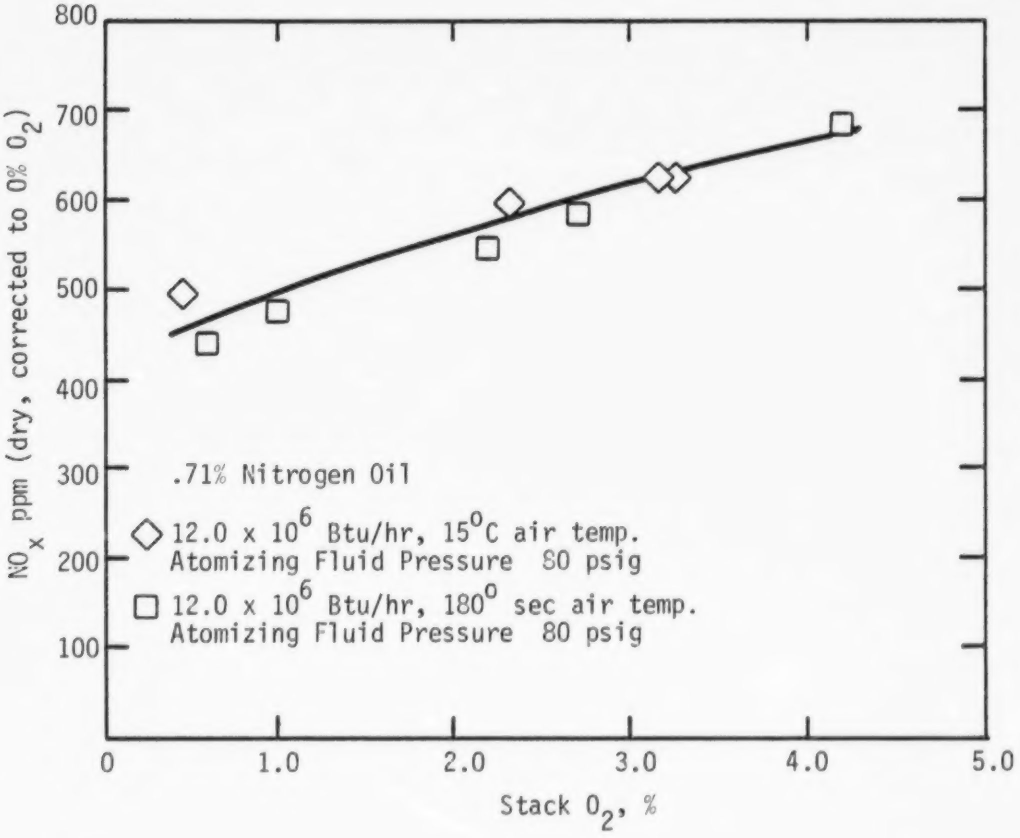


Figure 11-6. The Effect of Air Preheat on Unstaged NO_x Emissions.

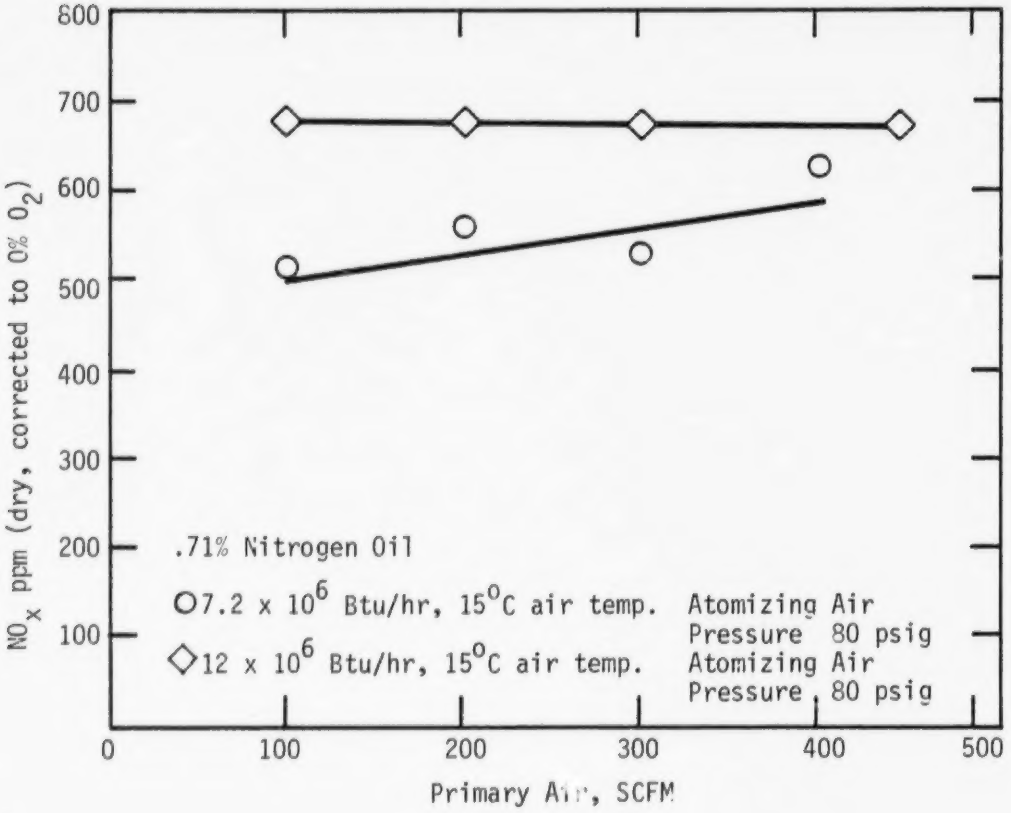


Figure 11-7. The Effect of Primary Air Flow Rate on Unstaged NO_x Emissions.

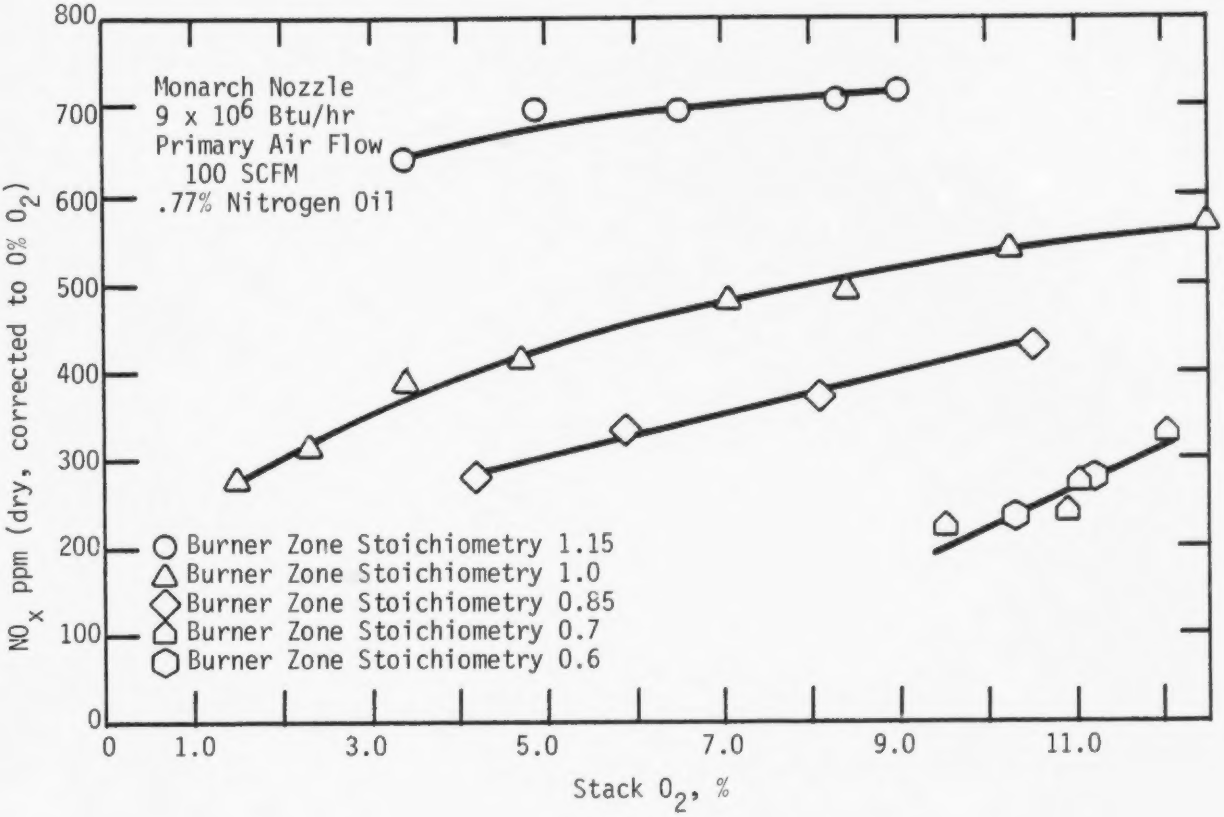


Figure 11-8. Effect of Primary Zone Stoichiometry on NO_x Production with the DMB.

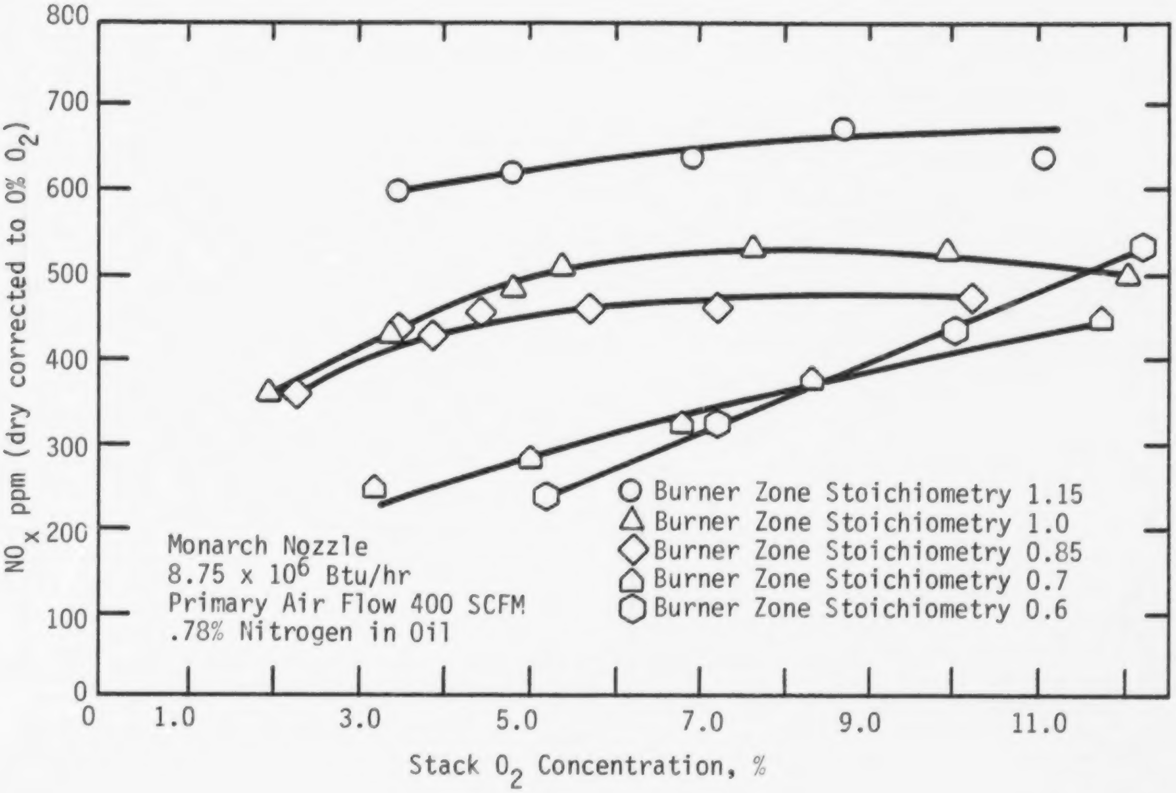


Figure 11-9. Effect of Primary Zone Stoichiometry on NO$_x$ Production with the DMB.

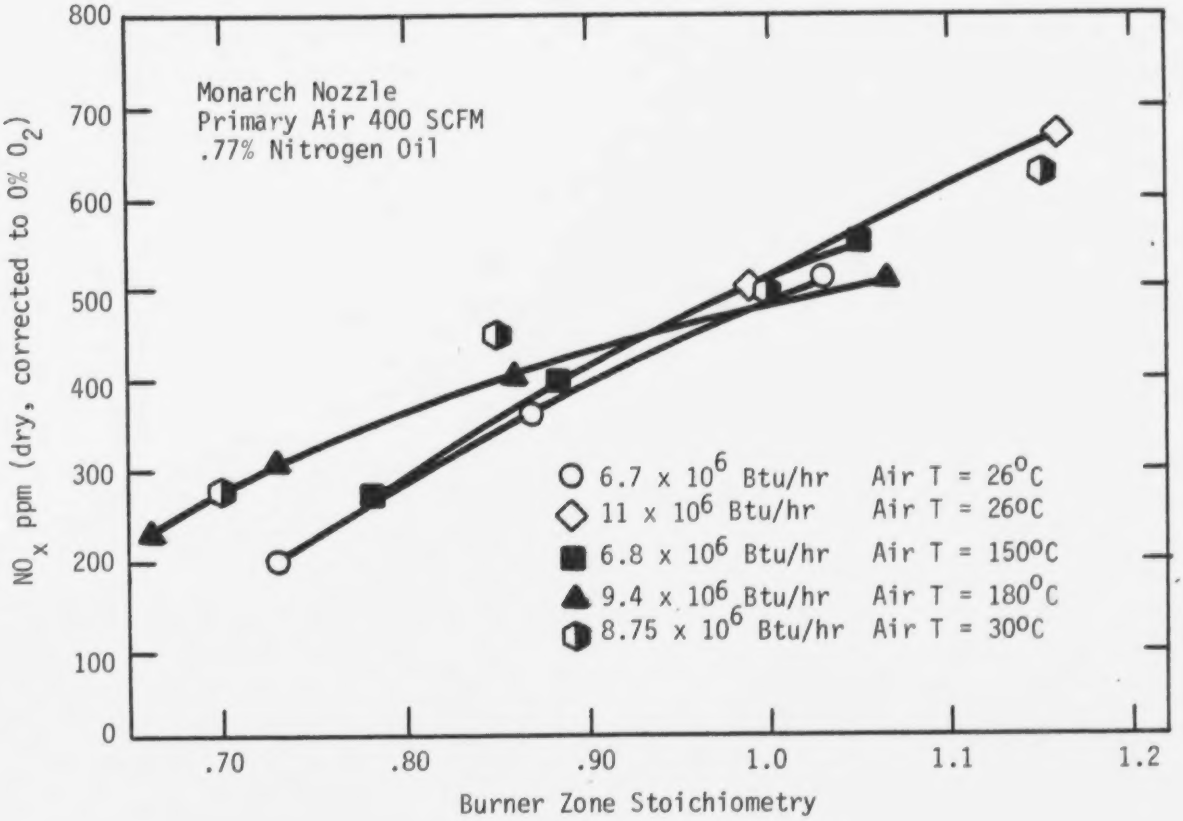


Figure 11-10. The Effect of Air Preheat on NO$_x$ Emission from the DMB (data interpolated to 5% overall excess oxygen).

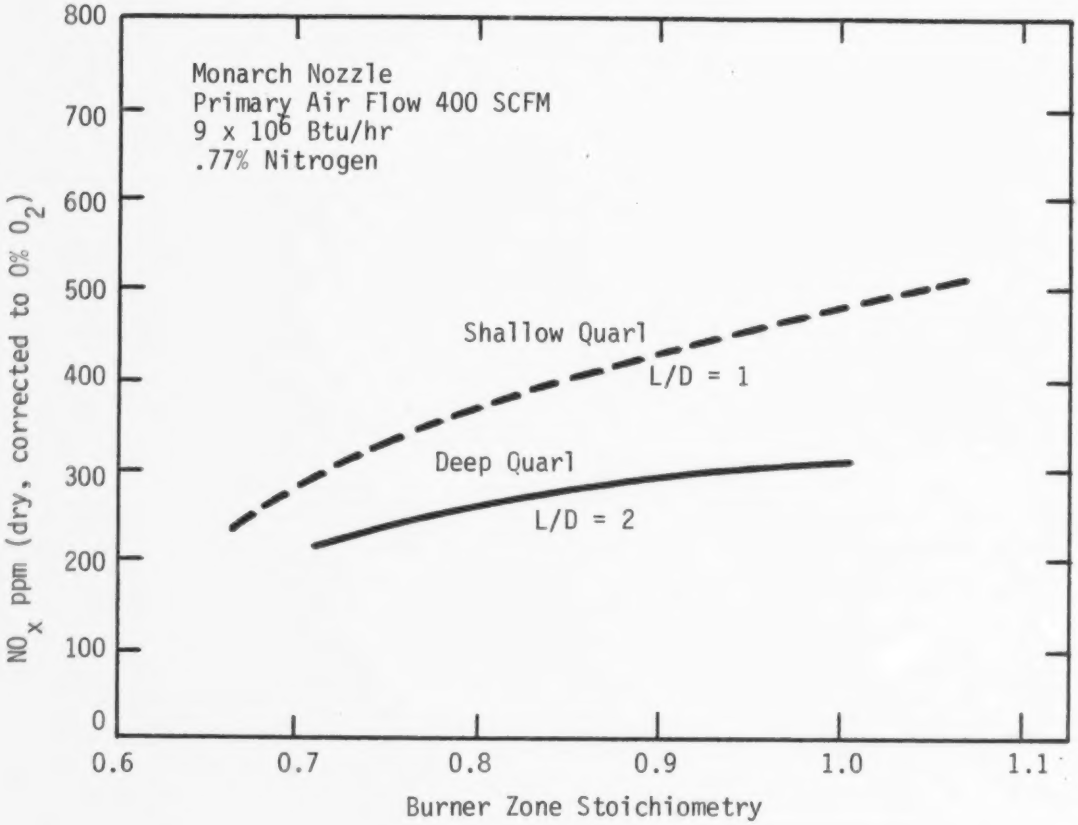


Figure 11-11. The Effect of Burner Exit Length on NO_x Emissions (data interpolated to 5% overall excess oxygen).

show the influence of burner zone stoichiometry upon NO production rate as a function of overall excess air for two primary air flow rates. Burner zone stoichiometry is calculated based upon the air flow through the primary and secondary channels (see Figure 11-2). At the lower primary air flow rates (100 SCFM), high staging levels could not be achieved without the emission of excessive amounts of smoke unless high overall excess air levels were maintained. Increasing the primary air flow rate to 400 SCFM allowed the burner to be operated at lower overall excess air levels with reduced primary zone stoichiometry (see Figure 11-9). Minimum emission levels of the order of 250 ppm were obtained with the Wilmington No. 6 fuel oil which has a nitrogen content of 0.76 percent.

The performance of the oil fired distributed mixing burner was not affected by air preheat (see Figure 11-10). Fuel air contacting in the early stages of the flame is strongly affected by the combustion air parameters, fuel injector conditions and burner geometry. Tests were carried out with two different refractory exits (quarls). The divergent angle for both was the same, 25^o, but the length was doubled for the deep quarl. The length to throat diameter ratio for the shallow and deep quarls was 1 and 2 respectively. Figure 11-11 compares NO_x emissions for the two exit geometries with a primary air flow of 400 SCFM. NO_x emissions were lowest with the deeper quarl. A test was carried out to assess the impact of outboard staged air distribution and velocity. Figure 11-12 compares emissions when the staged air is supplied through eight or sixteen discrete ports. NO_x emissions appear to be relatively insensitive to staged air injection parameters. A low nitrogen (0.24%) Indonesian/Malaysian oil was also tested. Figure 11-13 compares emissions for one set of conditions and the high and low nitrogen oils. Emissions with the low nitrogen oil were much less than those with the high nitrogen oil and they were almost insensitive to burner zone stoichiometry. In fact emissions reached a maximum with a burner zone stoichiometry of approximately 0.7. Screening tests indicated that variations in load, preheat primary air flow or atomizing fluid pressure had little impact on NO_x emissions.

11.3.2 Todd Nozzle

Unstaged Performance

The emissions characteristics of the DMB fitted with a Todd nozzle (8x0.105" round holes, 90^o spray angle) are defined, using an Indonesian/Malaysian residual fuel oil (0.24% N, 0.25% S). In general, flames were more unstable than

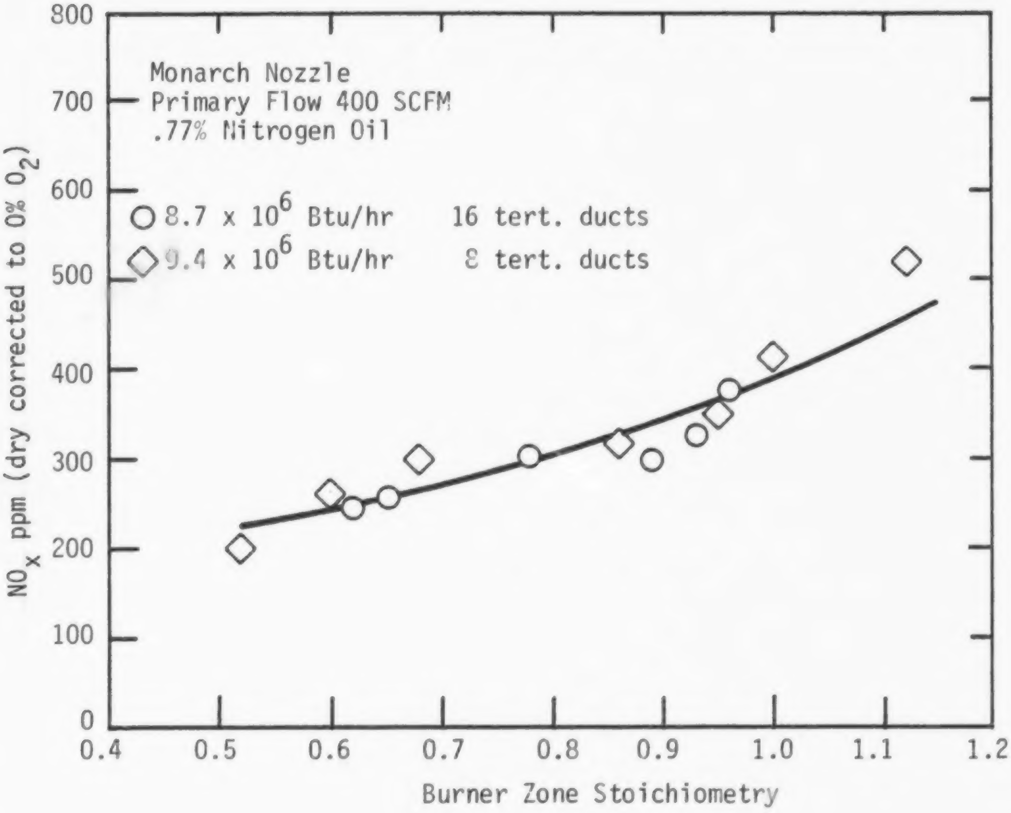


Figure 11-12. The Influence of Tertiary Air Velocity on Staged NO_x Emissions.

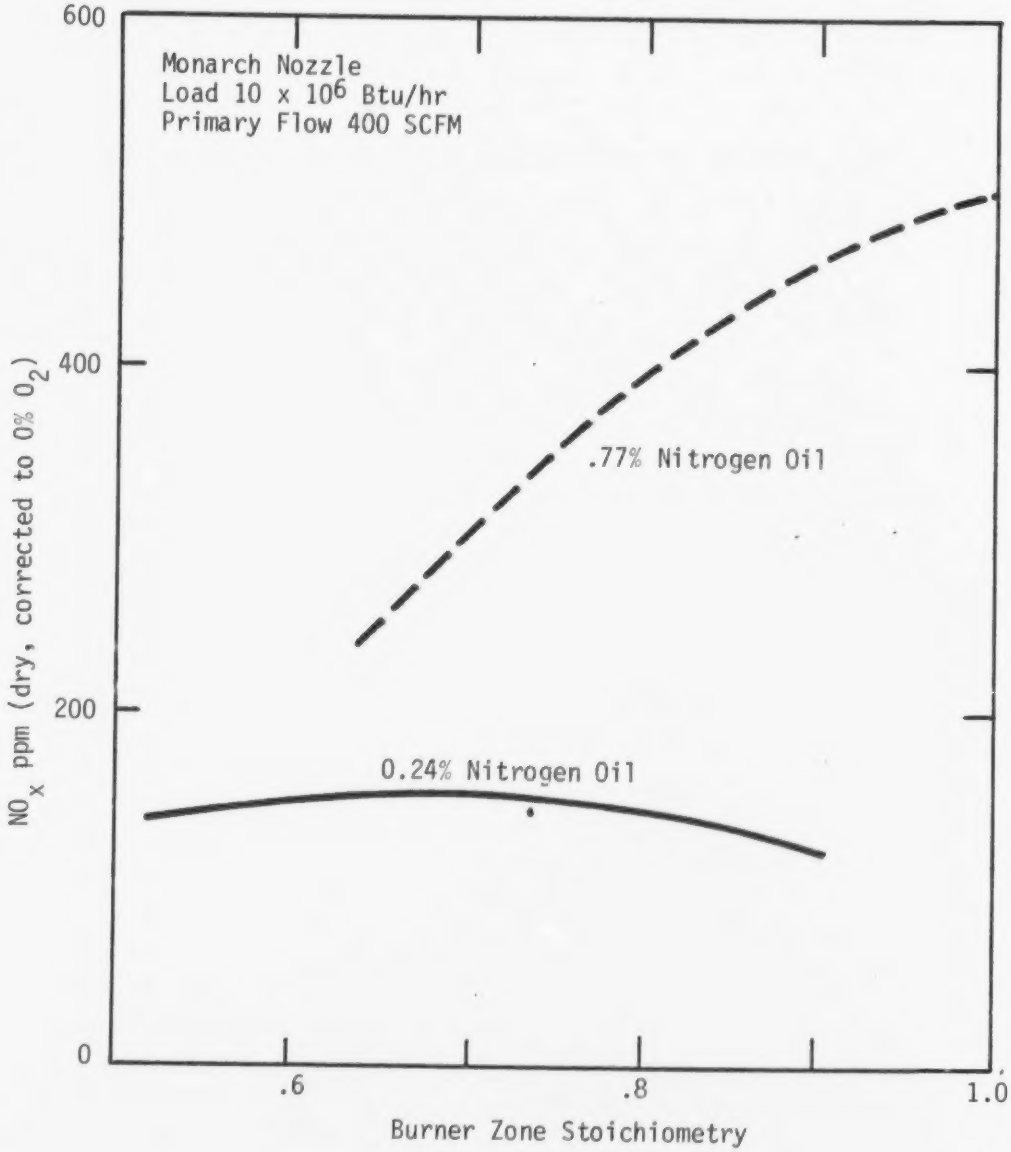


Figure 11-13. Comparison of Emission Performance of the Oil Fired DMB When Burning Oils of Different Nitrogen Contents (data interpolated to 5% excess oxygen).

that obtained from the 100 gph Monarch nozzle. Results from the Todd atomizer showed a slight reduction (20 ppm) in NO_x emission when atomizing pressure was increased from 53 psig to 70 psig (Figure 11-14). In addition, atomizing pressure of 70 psig produced a more unstable flame.

Figure 11-15 shows NO_x emissions at three primary air flow rates with 70 psig atomizing pressure. No difference was observed. With 53 psig atomizing pressure (see Figure 11-16), NO_x emissions were reduced by 25 ppm when the primary air was increased to an equivalent of 36% of the theoretical air requirement. Figure 11-17 illustrates the response of NO_x emission to primary air flow rate. The abrupt change at about 300 SCFM may be indicative of a transition in the aerodynamic interaction between the primary and secondary air streams. NO_x emissions were not affected by heating the combustion air from room temperature to 130^0C (see Figure 11-18) similarly to the results obtained with the Monarch nozzle.

Figure 11-19 presents data that indicates increasing the firing rate from 7×10^6 Btu/hr to 8.5×10^6 Btu/hr increased NO_x emission by about 100 ppm. Similar increase was observed with the .77% nitrogen fuel oil when firing rate was increased from 7×10^6 Btu/hr to 12×10^6 Btu/hr with the Monarch nozzle.

Staged Performance

With the low nitrogen oil (0.24%), emissions were relatively insensitive to staged operation. This was very similar to the performance with the Monarch nozzle and this oil. Figure 11-20 provides a comparison of NO_x emissions as a function of burner zone stoichiometry for the Todd and the Monarch nozzles with the 0.24% nitrogen oil.

11.4 Discussion and Conclusions

Figure 11-21 summarizes the performance of the 10×10^6 Btu/hr DMB with a 0.71% nitrogen fuel oil for short and long exit divergents under staged and unstaged conditions. The data was obtained with the Monarch nozzle. It can be seen that in both staged and unstaged the longer divergent gave lower emissions and that operating staged gave only slightly lower emissions than those obtained unstaged at low excess air. NO_x emissions appears to be a strong function of both excess air and burner zone equivalence ratio. Preliminary tests with a 50×10^6 Btu/hr prototype burner operating unstaged at two loads indicated similar characteristics (see Figure 11-22). Thus a scale increase of

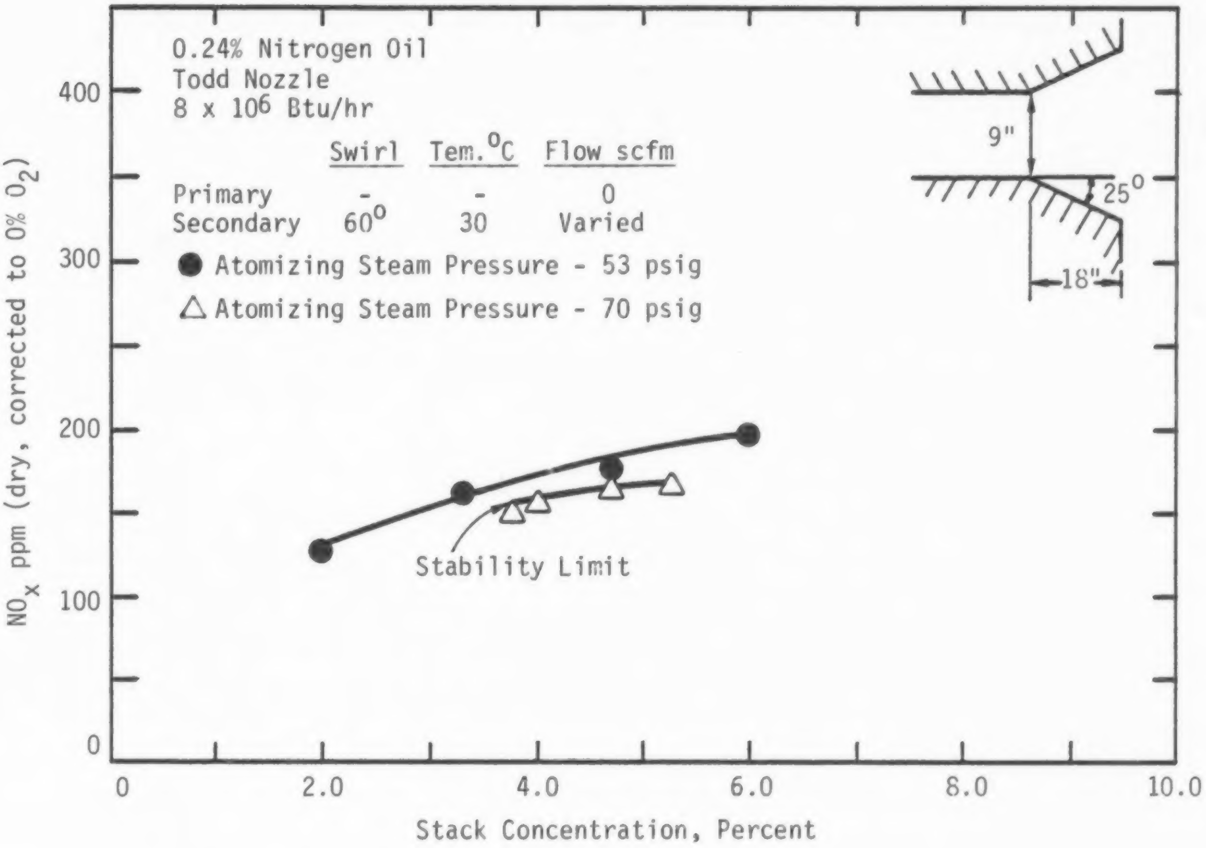


Figure 11-14. Effect of Atomizing Steam Pressure.

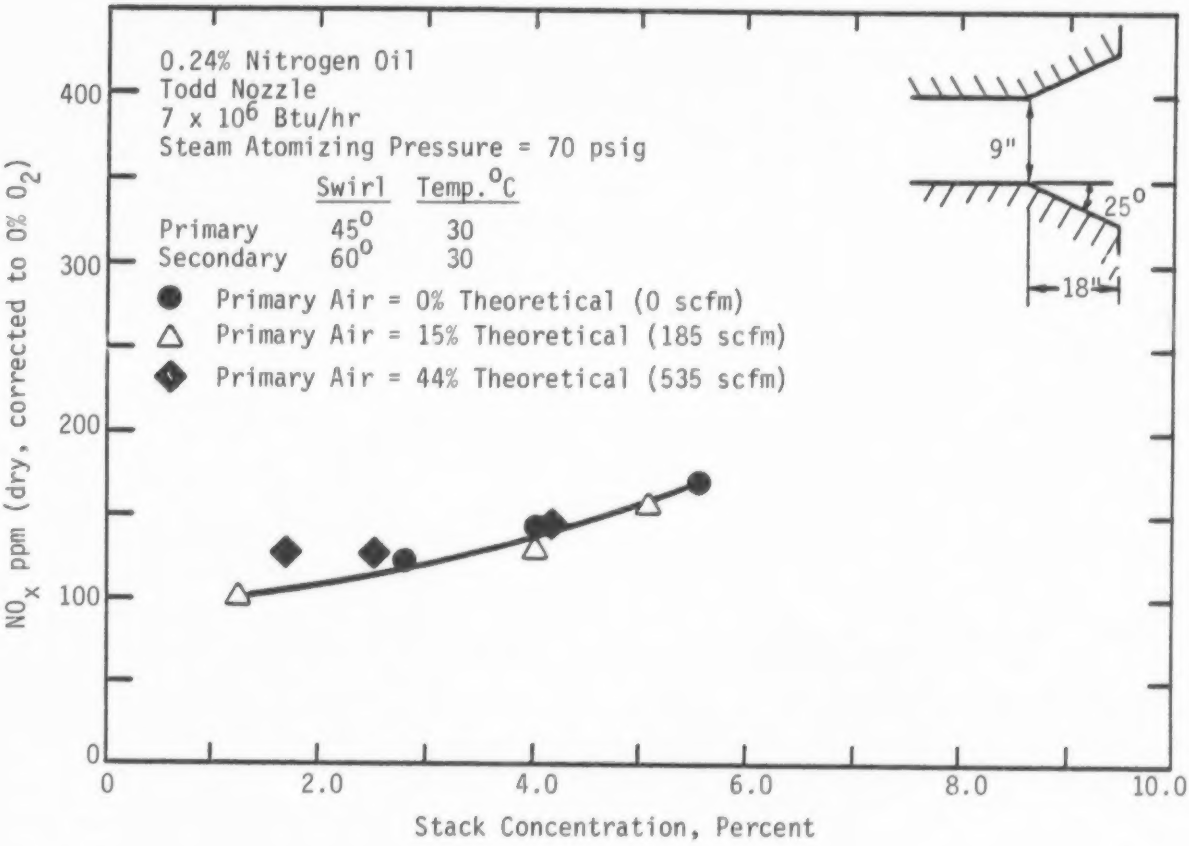


Figure 11-15. Effect of Primary Air (70 psig).

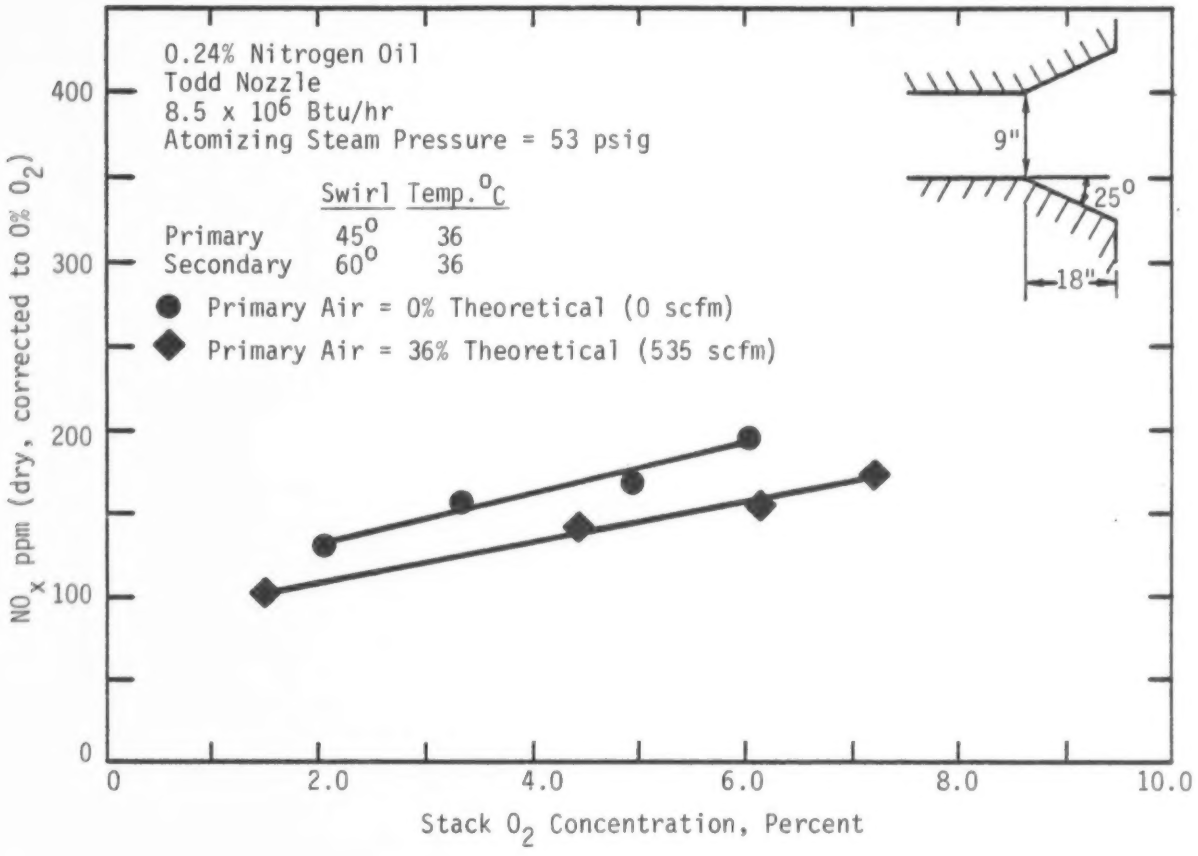


Figure 11-16. Effect of Primary Air (53 psig).

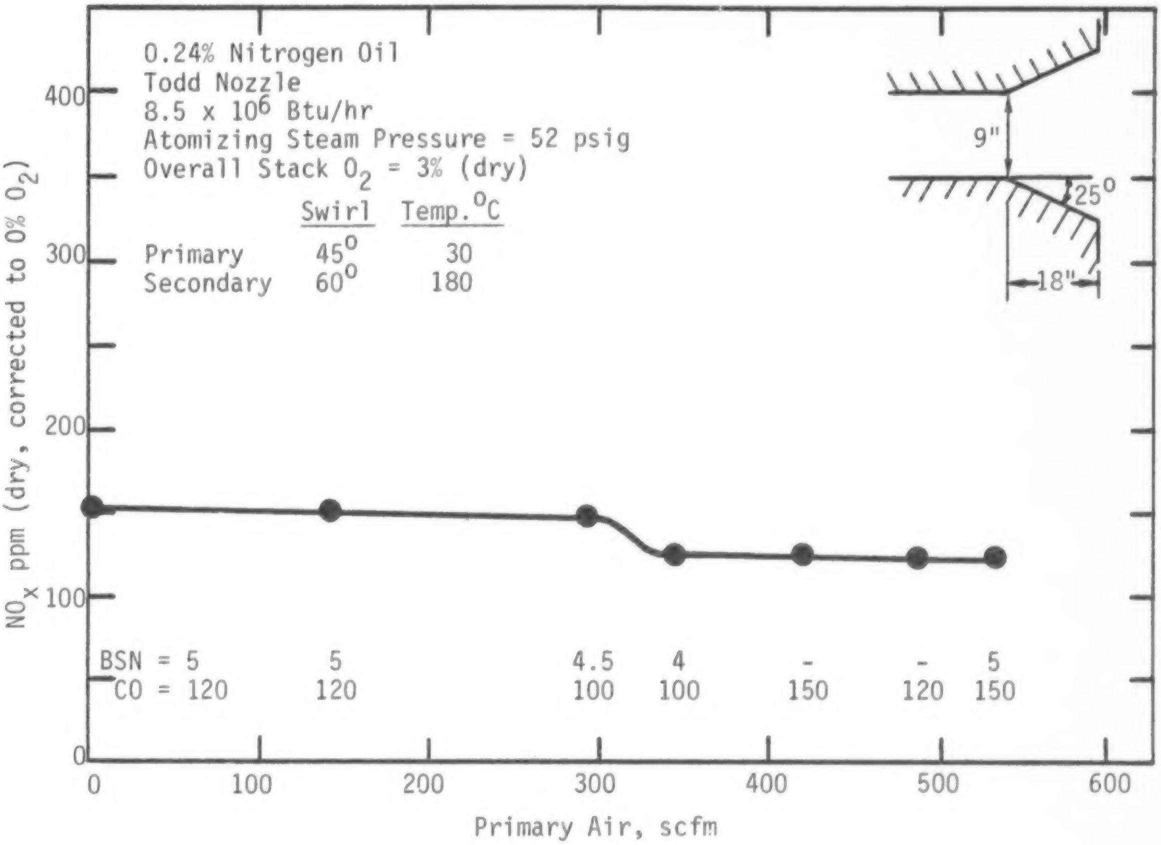


Figure 11-17. Primary Air Effect at 8.5 x 10^6 Btu/hr (52 psig).

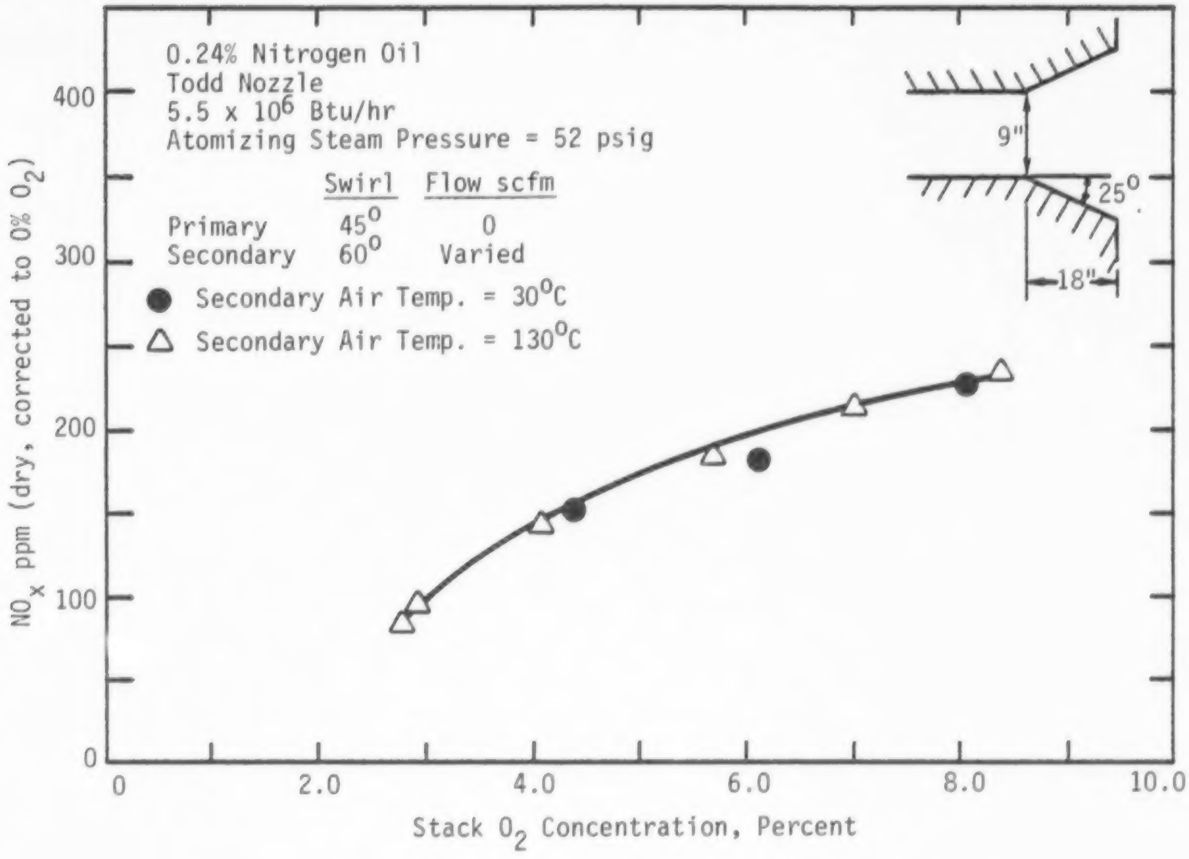


Figure 11-18. Effect of Air Preheat.

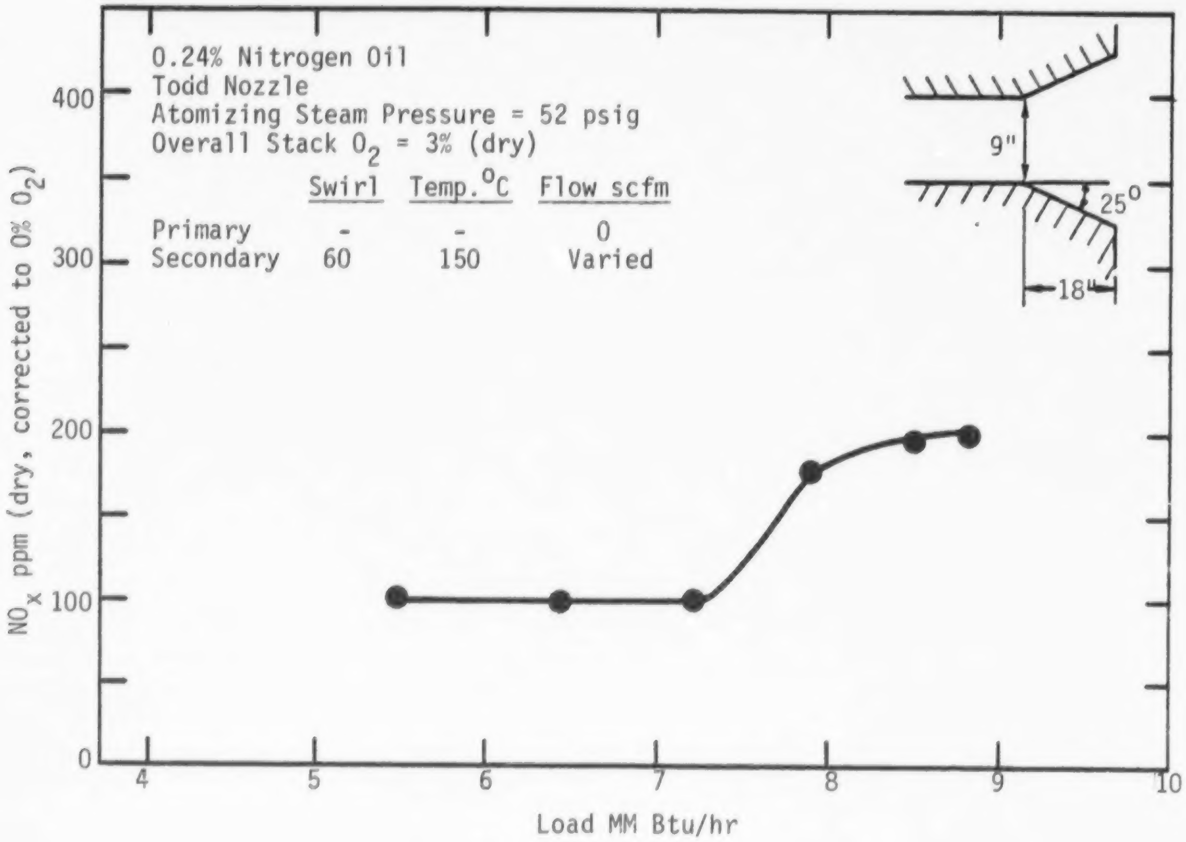


Figure 11-19. Effect of Firing Rate.

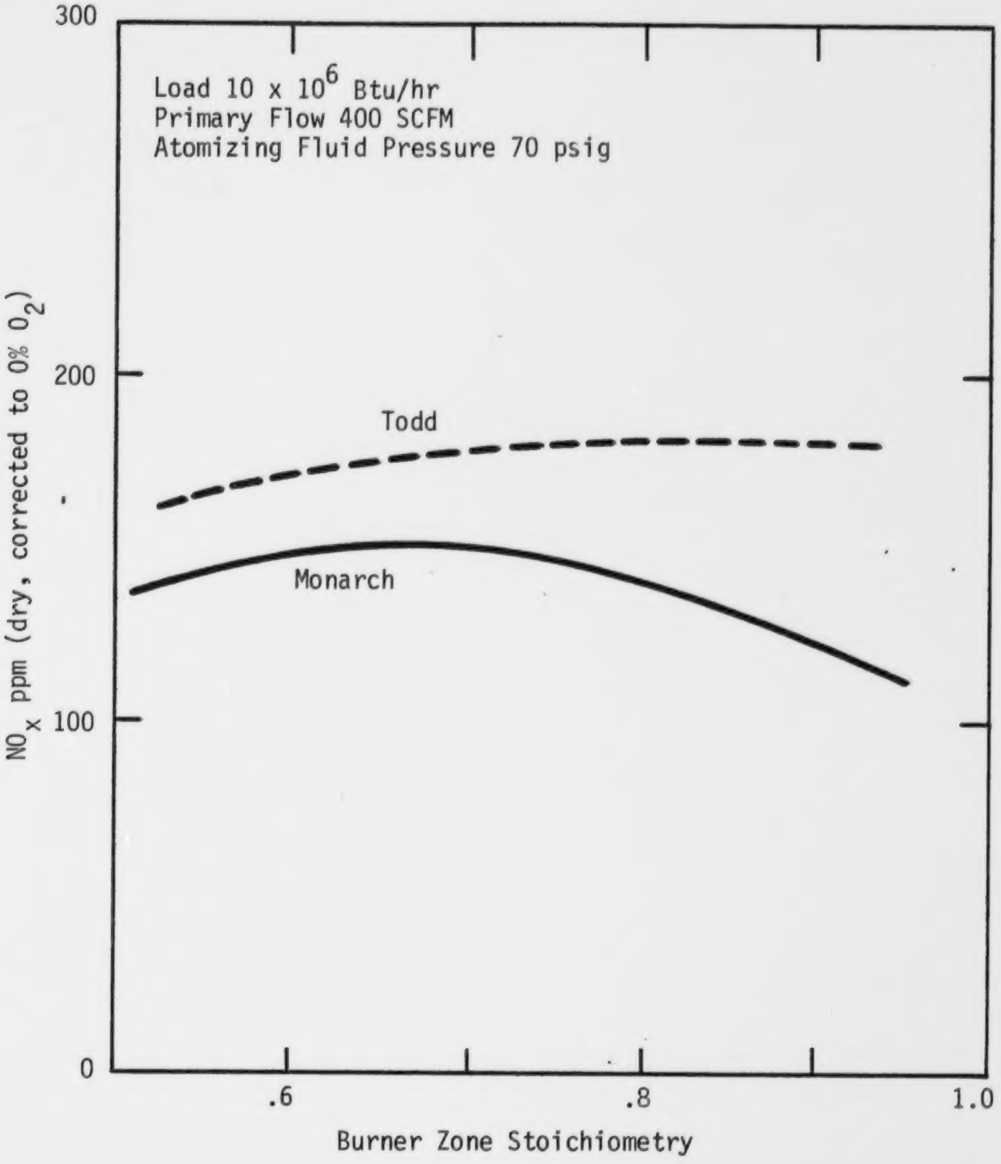


Figure 11-20. Comparison of Emission Performance with 0.24% Nitrogen Oil and Two Nozzles (data interpolated to 5% excess oxygen).

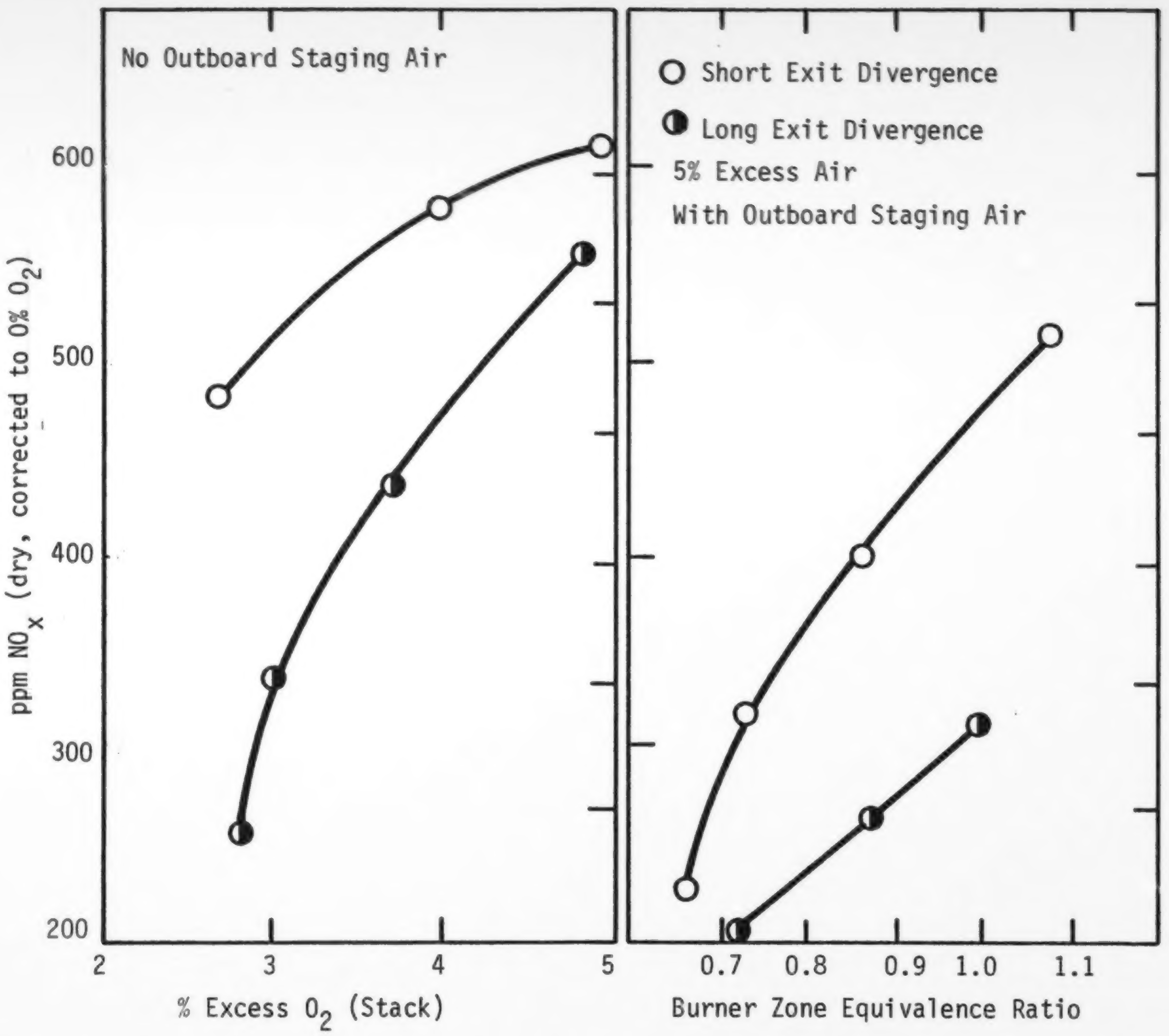


Figure 11-21. Emission Characteristics of a 10 x 10^6 Btu/hr DMB (Monarch nozzle, 70 psig steam pressure, 0.77% nitrogen oil).

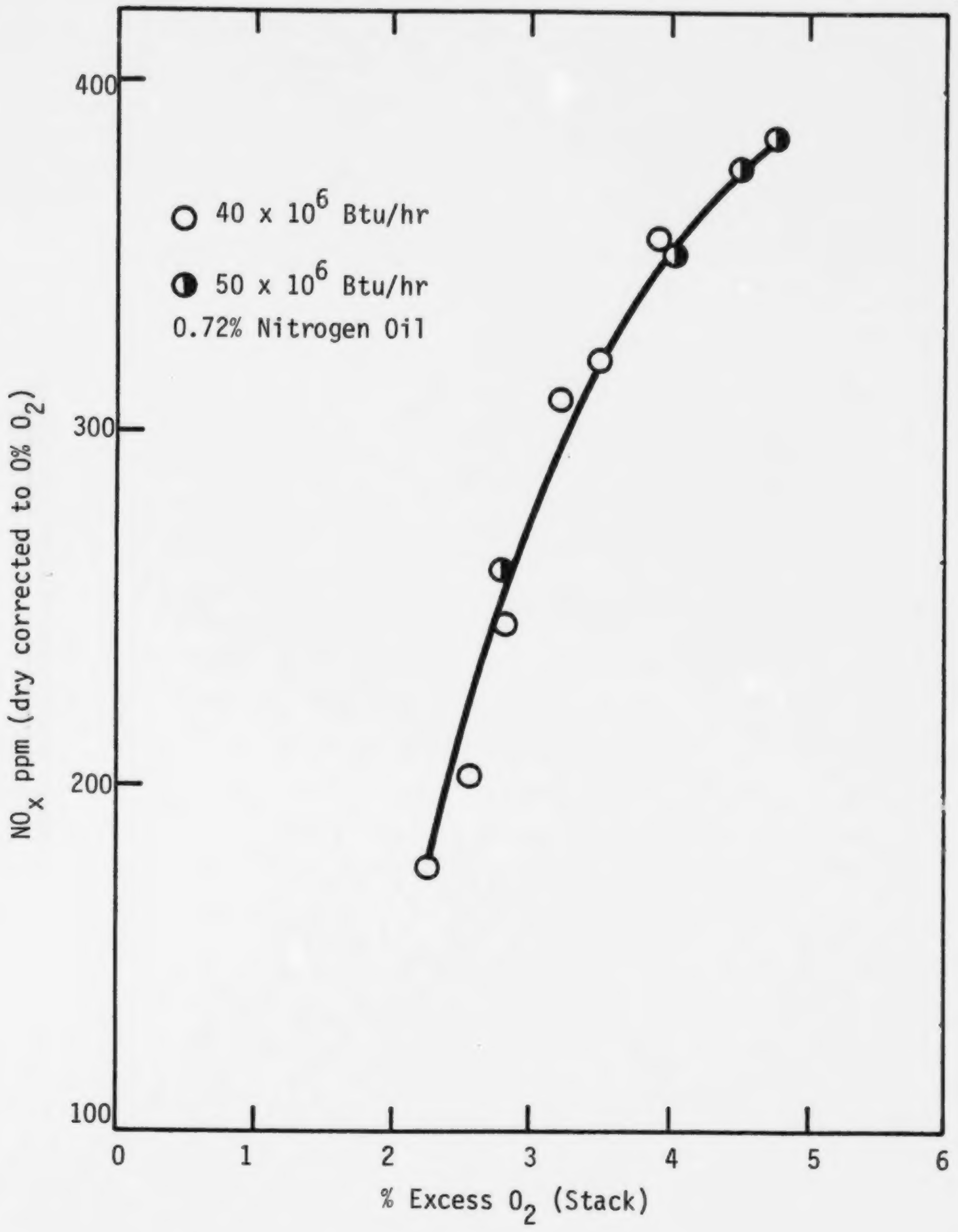


Figure 11-22. Unstaged DMB Performance (40 x 10^6 Btu/hr, 0.77% Nitrogen Oil).

four does not drastically affect performance.

Figure 11-23 compares the performance of the oil fired DMB, which is an aerdynamically staged device, with the low NO_x firetube boiler burner (68-02-1500), which is a physically staged device. Data is presented on the same figure which shows the range of emissions obtained by England et al(49) by optimizing the air flow distribution and the method of atomization. The data were obtained in a firetube simulator with a relatively high volumetric heat release rate compared to the water tube simulators. Figure 11-21 also shows the correlation by Cato et al(3) for industrial boiler emissions as a function of fuel nitrogen content. It can be seen that the most spectacular reductions in NO_x emissions using low NO_x burners will be achieved with high nitrogen fuels. It should be noted that neither the low NO_x firetube boiler burner nor the DMB appear capable of achieving NO_x emssions close to 100 ppm at 0% O_2 without further modification.

The major obstacle to reducing NO_x emissions further is the problem of carbonaceous particulate formation. Staged combustion achieved either physically or aerodynamically tends to increase particulate formation. Unless adequate precautions are taken, reduction in NO_x emissions from oil fired systems beyond 50% are usually accompanied by an increase in smoke formation. Further work is needed to understand soot formation in flames in order to alleviate this problem.

This preliminary study indicates that the DMB does have potential to reduce NO_x emissions associated with fuel oil firing. The benefits are most evident with high nitrogen fuels. Thus, this type of burner could be most suitable for burning coal or shale derived liquids which have high nitrogen contents. The relatively less impressive performance with low nitrogen fuels is due to the fact that the contribution of thermal NO_x to the total emission increase as the fuel nitrogen level decreases and the DMB is not designed specifically to limit thermal NO formation. It is unlikely that a burner of this type will ever be designed to produce very low levels of nitrogen oxides (less than 100 ppm for a 0.5% nitrogen oil) which could be possible with physically staged systems. The main problem is the increase of CO and smoke at some critical burner zone stoichiometry which prevents further reductions in NO_x emissions.

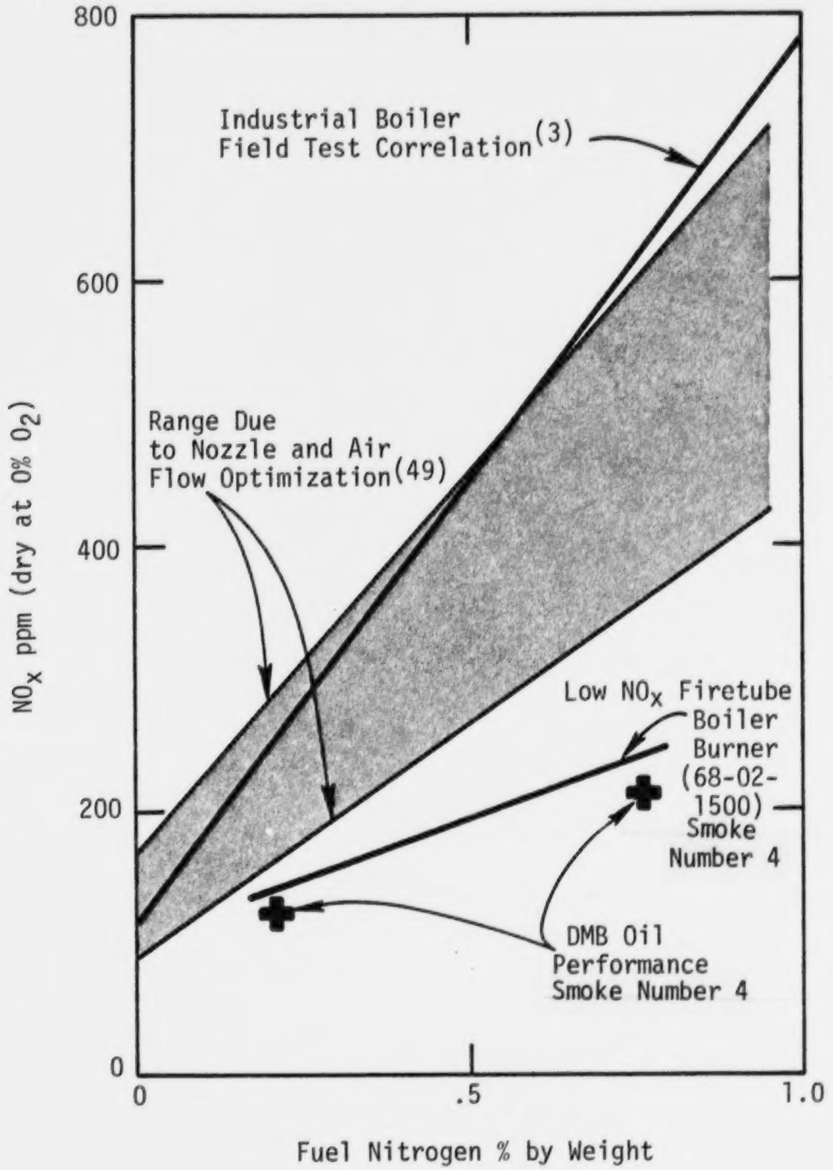


Figure 11-23. NO_x Control for Industrial Boilers Using Low NO_x Burners.

Although these preliminary studies with the oil-fired DMB were promising, particularly when firing high-nitrogen fuels, further efforts are needed to assess its true potential. A reduction in NO_x formation during the combustion of a nitrogen-containing fuel requires that the partially oxidized fuel reside for some time under oxygen-deficient conditions to maximize the formation of molecular nitrogen from the fuel nitrogen fragments. These conditions are conducive to soot formation, consequently it is possible that particulate emissions increase when NO levels are decreased. Recognizing that fuel-rich conditions are manditory to minimize NO_x emissions, an efficient low NO_x oil burner for liquid fuels should have a well-mixed fuel-rich zone at the optimum stoichiometry for NO_x reduction and an efficient second-stage mixing process to maximize burnout. The well mixed fuel-rich zone will preclude the formation of zones which are very fuel-rich and which will give rise to excessive soot production. Two design features can be optimized to improve the performance of the oil-fired DMB, these are:

- The Fuel Injection System - an efficient fuel injection system will promote mixing and distribute the fuel evenly. An injector which gives small droplets would probably be beneficial.
- Throat Velocities - the axial and tangential velocity at the burner throat control fuel-air mixing and therefore influence smoke emissions.

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APPENDIX A. FIELD TEST EXPERIMENTAL DATA

Table I. Field Test Experimental Data (1/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
3-74	Watertube	No. 5	EAV	W1-1	10,000	17	152	Steam,39	4.6	<10	-	139	177	NA	NA	415
"	"	"	"	W1-2	10,500	18	152	" 39	3.8	<10	-	118	144	"	"	415
"	"	"	"	W1-3	10,000	18	152	" 40	6.0	<10	-	142	199	"	"	435
"	"	"	"	W1-4	10,000	20	152	" 52	7.0	<10	-	155	232	"	"	425
"	"	"	"	W1-5	10,000	16	152	" 29	3.1	<10	-	93	109	"	"	420
"	"	"	"	W1-6	10,000	15	110	" 36	4.5	<10	-	117	148	"	"	410
"	"	"	"	W1-7	10,250	19	165	" 39	5.3	<10	11.0	140	187	"	"	415
"	"	"	"	W1-8	15,000	29	148	" 50	2.6	25	14.0	123	140	"	"	460
"	"	"	"	W1-9	15,000	32	147	" 52	8.1	<10	10.0	135	220	"	"	515
"	"	"	"	W1-10	15,500	32	147	" 52	6.1	<10	11.0	143	202	"	"	510
"	"	"	"	W1-11	15,000	32	150	" 51	4.2	<10	13.0	140	175	"	"	490
"	"	"	"	W1-12	15,000	30	150	" 50	2.7	100	14.0	110	126	"	"	470
"	"	"	"	W1-13	15,250	32	150	" 52	2.3	3000+	14.0	87	98	"	"	440
"	"	"	"	W1-14	15,000	31	150	" 51	2.8	3000	14.0	78	90	"	"	460
"	"	"	"	W1-15	15,000	34	150	" 63	6.2	<10	11.0	120	170	"	"	490
"	"	"	"	W1-16	15,000	30	150	" 50	4.6	<10	12.0	102	131	"	"	490
"	"	"	"	W1-17	19,000	41	151	" 61	4.3	<10	14.0	130	163	"	"	515
"	"	"	"	W1-18	18,500	41	150	" 60	9.3	<10	9.0	120	215	"	"	555
"	"	"	"	W1-19	18,500	42	149	" 61	7.6	<10	10.0	130	203	"	"	550
"	"	"	"	W1-20	18,500	42	149	" 60	5.5	<10	12.0	135	182	"	"	530
"	"	"	"	W1-21	18,500	40	150	" 60	3.6	100	14.0	110	132	"	"	510
"	"	"	"	W1-22	18,500	40	150	" 59	4.2	1150	12.0	88	110	"	"	500
"	"	"	"	W1-23	18,500	40	150	" 60	4.5	<10	13.0	117	148	"	"	520
"	"	"	"	W1-24	12,500	22	149	" 43	4.5	<10	13.0	98	124	"	"	450
"	"	"	"	W1-25	7,500	11	147	" 32	8.4	<10	9.0	107	178	"	"	405
"	"	"	"	W1-26	8,750	16	149	" 37	3.2	2500	15.0	66	78	"	"	405
"	"	"	"	W1-27	14,500	28	152	" 48	3.1	<10	-	113	133	"	"	455

Table I. Field Test Experimental Date (2/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
3-74	Watertube	No. 5	RO, 69%	W1-28	14,750	29	152	Steam,49	1.5	3000+	-	105	113	NA	NA	445
"	"	"	" 75%	W1-29	14,500	28	152	" 49	1.8	2200	-	110	120	"	"	445
"	"	"	" 81%	W1-30	14,500	28	152	" 48	2.3	500	-	115	129	"	"	450
"	"	"	" 88%	W1-31	14,500	27	152	" 47	2.1	<10	-	118	131	"	"	450
"	"	"	" 94%	W1-32	14,500	27	150	" 47	2.4	25	-	120	135	"	"	460
"	"	"	" 100%	W1-33	14,500	27	152	" 48	2.6	<10	-	120	137	"	"	460
"	"	Natural Gas	EAV	W1-34	13,500	-	-	NA	2.5	30	-	62	70	"	"	430
"	"	"	"	W1-35	13,000	-	-	"	6.2	15	-	55	78	"	"	440
"	"	"	"	W1-36	13,000	-	-	"	4.5	-	-	62	79	"	"	440
"	"	"	"	W1-37	13,000	-	-	"	1.0	700	-	59	62	"	"	430
"	"	"	"	W1-38	10,000	-	-	"	4.0	15	-	62	77	"	"	410
"	"	"	"	W1-39	10,250	-	-	"	6.3	30	-	57	81	"	"	415
"	"	"	"	W1-40	10,500	-	-	"	4.7	30	-	62	80	"	"	415
"	"	"	"	W1-41	10,500	-	-	"	2.3	50	-	63	71	"	"	410
"	"	"	RO,100%	W1-42	13,000	-	-	"	2.5	30	-	64	72	"	"	430
"	"	"	" 94%	W1-43	13,500	-	-	"	2.8	15	-	67	77	"	"	435
"	"	"	" 88%	W1-44	13,500	-	-	"	2.9	15	-	70	81	"	"	435
"	"	"	" 81%	W1-45	13,000	-	-	"	2.4	15	-	73	82	"	"	435
"	"	"	" 75%	W1-46	12,750	-	-	"	1.1	420	-	77	81	"	"	430
"	"	"	" 100%	W1-47	12,500	-	-	"	2.5	30	-	64	72	"	"	425
"	"	"	EAV	W1-48	20,000	-	-	"	7.6	20	-	50	78	"	"	410
"	"	"	"	W1-49	20,000	-	-	"	10.3	15	-	38	75	"	"	525
"	"	"	"	W1-50	19,750	-	-	"	9.4	20	-	44	79	"	"	525
"	"	"	"	W1-51	19,500	-	-	"	6.1	30	-	54	76	"	"	500
"	"	"	"	W1-52	19,250	-	-	"	5.0	785	-	53	69	"	"	490
"	"	"	"	W1-53	15,750	-	-	"	6.3	5	-	51	72	"	"	460
"	"	"	"	W1-54	16,000	-	-	"	8.9	5	-	44	76	"	"	475

Table I. Field Test Experimental Data (3/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
3-74	Watertube	Natural Gas	EAV	W1-55	16,000	-	-	NA	7.6	20	-	49	77	NA	NA	475
"	"	"	"	W1-56	15,500	-	-	"	4.7	1000+	-	50	64	"	"	460
"	"	"	"	W1-57	8,250	-	-	"	8.5	<10	-	44	74	"	"	400
"	"	"	"	W1-58	8,000	-	-	"	10.6	<10	-	39	79	"	"	395
"	"	"	"	W1-59	8,500	-	-	"	9.4	<10	-	42	76	"	"	395
"	"	"	"	W1-60	8,500	-	-	"	6.3	<10	-	48	68	"	"	395
"	"	"	"	W1-61	8,500	-	-	"	4.5	470	-	47	60	"	"	395
"	Firetube	No. 5	EAV	F1-1	4,000	-	-	Air, 39	5.5	<10	-	170	230	"	"	330
"	"	"	"	F1-2	5,000	34	150	" 39	2.0	<10	-	180	199	"	"	330
"	"	"	"	F1-3	5,500	34	155	" 37	0.9	120	-	170	178	"	"	330
"	"	"	"	F1-4	5,500	34	155	" 37	4.2	<10	-	180	228	"	"	330
"	"	"	"	F1-5	5,000	34	150	" 37	6.1	<10	-	170	244	"	"	340
"	"	"	"	F1-6	7,500	42	155	" 43	7.4	<10	-	165	257	"	"	355
"	"	"	"	F1-7	7,250	42	155	" 43	8.7	<10	-	155	268	"	"	360
"	"	"	"	F1-8	7,500	40	155	" 42	5.9	<10	-	173	241	"	"	350
"	"	"	"	F1-9	7,500	42	155	" 42	3.2	<10	-	175	206	"	"	350
"	"	"	"	F1-10	7,500	42	160	" 52	0.8	1000+	-	163	171	"	"	370
"	"	"	"	F1-11	11,000	58	160	" 52	3.7	<10	-	193	237	"	"	375
"	"	"	"	F1-12	11,500	58	160	" 53	2.4	<10	-	190	215	"	"	370
"	"	"	"	F1-13	11,000	54	160	" 51	3.9	<10	-	202	250	"	"	375
"	"	"	"	F1-14	11,000	56	140	" 51	3.5	<10	-	197	239	"	"	375
"	"	"	"	F1-15	8,000	45	150	" 45	6.5	<10	-	175	260	"	"	365
"	"	"	"	F1-16	11,000	56	140	" 50	3.2	<10	-	195	231	"	"	375
"	"	Natural Gas	"	F1-17	1,000	-	-	NA	13.3	20	-	42	115	"	"	330
"	"	"	"	F1-18	1,000	-	-	"	10.4	20	-	44	87	"	"	330
"	"	"	"	F1-19	1,000	-	-	"	9.6	20	-	47	87	"	"	330
"	"	"	"	F1-20	750	-	-	"	14.9	20	-	38	130	"	"	330

Table I. Field Test Experimental Data (4/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
3-74	Firetube	Natural Gas	EAV	F1-21	3,750	-	-	NA	9.1	15	-	61	107	NA	NA	340
"	"	"	"	F1-22	3,750	-	-	"	8.0	15	-	64	103	"	"	340
"	"	"	"	F1-23	4,000	-	-	"	10.7	17	-	59	120	"	"	345
"	"	"	"	F1-24	3,875	-	-	"	11.5	5	-	56	123	"	"	345
"	"	"	"	F1-25	7,000	-	-	"	6.5	5	-	73	105	"	"	355
"	"	"	"	F1-26	6,500	-	-	"	7.6	15	-	71	111	"	"	360
"	"	"	"	F1-27	6,500	-	-	"	9.0	5	-	68	119	"	"	360
"	"	"	"	F1-28	8,500	-	-	"	7.1	15	-	68	102	"	"	360
"	"	"	"	F1-29	8,000	-	-	"	6.6	40	-	72	105	"	"	365
"	"	"	"	F1-30	8,750	-	-	-	7.8	40	-	66	105	"	"	370
12-74	Firetube	No.5	EAV,FGR	F2-1	4,000	33	141	Air,36	1.5	<10	-	167	180	0	NA	330
"	"	"	"	F2-2	4,100	33	142	" 36	4.3	<10	-	162	204	0	"	330
"	"	"	"	F2-3	4,000	33	142	" 36	4.0	<10	-	160	198	23	"	335
"	"	"	"	F2-4	4,000	33	143	" 36	4.0	<10	-	145	179	32	"	340
"	"	"	"	F2-5	4,000	33	144	" 36	4.0	<10	-	118	148	40	"	340
"	"	"	"	F2-6	3,900	33	145	" 36	5.9	<10	-	160	223	0	"	340
"	"	"	"	F2-7	4,000	33	145	" 36	5.7	<10	-	152	209	23	"	340
"	"	"	"	F2-8	3,900	33	145	" 36	5.6	<10	-	129	176	32	"	345
"	"	"	"	F2-9	4,100	33	145	" 36	5.9	<10	-	123	171	39	"	350
"	"	"	"	F2-10	4,200	33	145	" 36	2.3	<10	-	175	197	0	"	335
"	"	"	"	F2-11	4,200	33	145	" 36	2.2	<10	-	168	188	19	"	335
"	"	"	"	F2-12	4,300	33	145	" 36	2.2	<10	-	153	171	28	"	335
"	"	"	"	F2-13	4,000	33	145	" 36	2.0	<10	-	139	154	39	"	335
"	"	"	"	F2-14	3,800	33	145	" 36	2.2	<10	-	118	132	53	"	345
"	"	"	"	F2-15	6,000	33	147	" 42	7.4	<10	-	145	224	0	"	360
"	"	"	"	F2-16	6,000	42	147	" 42	7.1	40	-	134	203	10	"	360

Table I. Field Test Experimental Data (5/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
12-74	Firetube	No.5	EAV,FGR	F2-17	6,000	40	147	Air,42	7.1	40	-	123	186	20	NA	360
"	"	"	"	F2-18	6,400	40	147	" 42	7.0	45	-	113	170	40	"	365
"	"	"	"	F2-19	6,500	40	147	" 42	6.2	40	-	120	171	29	"	365
"	"	"	"	F2-20	6,200	40	148	" 42	4.2	40	-	120	150	35	"	360
"	"	"	"	F2-21	6,300	39	148	" 40	4.0	40	-	123	152	28	"	360
"	"	"	"	F2-22	6,300	40	149	" 42	4.2	40	-	133	166	18	"	355
"	"	"	"	F2-23	6,300	39	148	" 41	4.1	40	-	162	202	0	"	350
"	"	"	"	F2-24	11,250	58	146	" 52	4.1	35	-	173	215	0	"	375
"	"	"	"	F2-25	11,000	57	147	" 52	3.0	35	-	171	200	0	"	370
"	"	"	"	F2-26	11,000	58	147	" 52	3.1	50	-	145	170	9	"	380
"	"	"	"	F2-27	11,000	57	146	" 51	2.5	58	-	127	170	18	"	380
"	"	"	"	F2-28	11,000	57	147	" 52	1.5	225	-	119	128	22	"	380
"	"	"	"	F2-29	10,500	57	146	" 51	2.0	78	-	125	138	20	"	380
"	"	"	"	F2-30	10,750	58	147	" 52	1.9	85	-	135	149	20	"	380
"	"	"	"	F2-31	11,250	57	146	" 52	2.3	55	-	141	158	10	"	-
1-75	"	"	"	F2-32	9,250	51	148	" 48	5.9	56	-	161	224	0	"	370
"	"	"	"	F2-33	9,000	51	148	" 48	5.8	45	-	140	194	9	"	375
"	"	"	"	F2-34	9,000	54	150	" 49	4.7	60	-	130	168	17	"	380
"	"	"	"	F2-35	9,800	52	150	" 48	3.5	70	-	125	150	26	"	380
"	"	"	"	F2-36	10,000	52	150	" 48	3.4	70	-	127	152	19	"	375
"	"	"	"	F2-37	9,900	50	150	" 48	3.5	70	-	141	169	10	"	370
"	"	"	"	F2-38	9,900	50	150	" 47	3.5	70	-	162	195	0	"	-
1-75	Watertube	No.5	EAV,FGR	W2-1	9,500	16	143	Steam,37	3.3	10	-	94	112	0	NA	380
"	"	"	"	W2-2	10,000	17	143	" 38	3.4	38	-	99	118	10	"	-
"	"	"	"	W2-3	10,000	17	144	" 38	3.4	52	-	95	113	19	"	410
"	"	"	"	W2-4	10,500	24	144	" 33	3.0	53	-	93	109	22	"	420

Table I. Field Test Experimental Data (6/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
1-75	Watertube	No.5	EAV,FGR	W2-5	10,000	19	143	Steam,40	2.1	110	-	82	91	22	NA	413
"	"	"	"	W2-6	10,200	20	143	" 40	1.9	112	-	78	86	27	"	420
"	"	"	"	W2-7	10,200	20	144	" 40	2.1	110	-	82	91	24	"	420
"	"	"	"	W2-8	9,800	19	144	" 40	2.1	110	-	84	93	20	"	415
"	"	"	"	W2-9	10,200	20	144	" 41	2.0	111	-	82	91	9	"	410
"	"	"	"	W2-10	10,300	20	144	" 41	2.1	210	-	83	92	0	"	410
"	"	"	"	W2-11	6,500	13	145	" 34	4.7	30	-	110	142	0	"	380
"	"	"	"	W2-12	6,700	14	145	" 36	4.7	27	-	102	132	11	"	380
"	"	"	"	W2-13	6,500	12	141	" 33	4.6	30	-	96	123	17	"	360
"	"	"	"	W2-14	7,000	12	140	" 33	4.6	30	-	91	117	24	"	370
"	"	"	"	W2-15	6,500	14	143	" 35	3.0	37	-	89	104	0	"	380
"	"	"	"	W2-16	6,500	14	143	" 35	2.8	30	-	92	106	9	"	385
"	"	"	"	W2-17	6,000	14	142	" 34	3.1	30	-	89	105	17	"	380
"	"	"	"	W2-18	6,400	14	140	" 35	2.7	30	-	84	96	27	"	380
"	"	"	"	W2-19	6,500	14	139	" 35	2.8	30	-	80	92	33	"	385
"	"	"	"	W2-20	4,800	9	139	" 30	8.8	30	-	91	157	0	"	390
"	"	"	"	W2-21	14,600	33	145	" 42	4.0	20	-	112	139	0	"	450
"	"	"	"	W2-22	16,200	39	144	" 57	4.2	20	-	113	141	8	"	470
"	"	"	"	W2-23	15,000	35	144	" 54	3.9	20	-	106	130	19	"	475
"	"	"	"	W2-24	14,900	34	144	" 53	4.2	20	-	108	135	24	"	475
"	"	"	"	W2-25	15,000	34	144	" 54	3.4	20	-	113	135	8	"	470
"	"	"	"	W2-26	15,500	36	144	" 55	5.4	20	-	110	148	9	"	470
"	"	"	"	W2-27	15,800	37	144	" 56	2.0	47	-	97	107	11	"	465
"	"	"	"	W2-28	15,600	35	144	" 55	1.9	365	-	91	100	0	"	460
"	"	"	"	W2-29	15,500	36	149	" 56	2.9	22	-	105	122	0	"	460
"	"	"	"	W2-30	16,000	36	143	" 56	4.0	20	-	114	141	0	"	463

Table I. Field Test Experimental Data (7/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
1-75	Watertube	No.5	EAV,FGR	W2-31	15,800	36	141	Steam,55	2.9	27	-	106	123	8	NA	473
"	"	"	"	W2-32	15,800	35	141	" 55	3.0	20	-	103	120	11	"	480
"	"	"	"	W2-33	16,000	36	141	" 55	3.1	20	-	101	119	21	"	482
"	"	"	"	W2-34	15,600	35	141	" 55	2.4	20	-	90	102	28	"	490
"	"	"	"	W2-35	16,300	38	140	" 57	4.3	15	-	103	130	24	"	500
"	"	"	"	W2-36	16,000	37	140	" 56	3.0	20	-	100	117	17	"	500
"	"	"	"	W2-37	16,200	37	140	" 57	4.2	20	-	113	141	16	"	500
"	"	"	"	W2-38	16,800	38	140	" 57	2.0	45	-	92	102	19	"	490
"	"	"	"	W2-39	16,800	39	140	" 58	4.5	20	-	112	143	7	"	490
"	"	"	"	W2-40	16,500	38	140	" 59	1.9	32	-	102	112	9	"	480
"	"	"	"	W2-41	16,700	39	140	" 59	4.4	20	-	115	146	0	"	480
"	"	"	"	W2-42	16,000	39	140	" 59	1.9	213	-	89	98	0	"	470
"	"	"	"	W2-43	16,000	39	140	" 59	3.4	<10	-	110	131	0	"	-
"	"	"	"	W2-44	15,200	39	140	" 59	5.0	<10	-	113	149	0	"	-
"	"	"	"	W2-45	15,500	39	140	" 59	2.1	<10	-	93	103	0	"	-
3-75	Firetube	No.5	CAS	F3-1	4,000	36	140	Air,37	6.8	20	-	129	191	NA	133	340
"	"	"	"	F3-2	6,000	34	140	" 36	7.3	20	-	115	176	"	117	340
"	"	"	"	F3-3	6,000	33	143	" 36	6.8	20	-	115	170	"	109	340
"	"	"	"	F3-4	7,000	33	142	" 36	5.4	30	-	113	152	"	96	340
"	"	"	"	F3-5	6,000	32	137	" 36	6.6	30	-	125	182	"	128	340
"	"	"	"	F3-6	6,000	32	148	" 36	5.3	30	-	116	155	"	97	340
"	"	"	"	F3-7	6,000	33	145	" 36	6.3	30	-	110	157	"	96	340
"	"	"	"	F3-8	6,000	33	145	" 36	5.1	40	-	100	132	"	85	340
"	"	"	"	F3-9	6,000	33	142	" 36	5.9	30	-	97	135	"	83	340
"	"	"	"	F3-10	6,000	33	145	" 36	5.2	40	-	90	120	"	74	340

Table I. Field Test Experimental Data (8/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
3-75	Firetube	No.5	CAS	F3-11	10,000	58	139	Air,51	3.6	50	-	172	208	NA	120	360
"	"	"	"	F3-12	13,000	58	140	" 52	4.8	50	-	157	204	"	127	355
"	"	"	"	F3-13	11,000	58	137	" 52	4.0	50	-	170	210	"	122	365
"	"	"	"	F3-14	11,500	58	135	" 52	5.1	50	-	155	205	"	111	365
"	"	"	"	F3-15	11,000	56	134	" 48	5.7	60	-	150	206	"	115	370
"	"	"	"	F3-16	7,000	46	135	" 45	8.0	60	-	130	210	"	128	360
"	"	"	"	F3-17	8,000	38	145	" 40	8.9	50	-	116	201	"	143	355
"	"	"	"	F3-18	7,000	40	142	" 40	5.0	50	-	126	165	"	98	350
"	"	"	"	F3-19	8,000	38	150	" 39	6.5	52	-	119	172	"	101	340
"	"	"	"	F3-20	7,100	38	152	" 39	7.5	50	-	112	145	"	94	350
"	"	"	"	F3-21	7,000	37	155	" 38	8.0	50	-	106	171	"	89	350
"	"	"	"	F3-22	7,100	38	150	" 39	8.4	50	-	104	173	"	93	350
"	"	"	"	F3-23	7,000	38	150	" 38	8.3	50	-	103	170	"	91	350
"	"	"	"	F3-24	5,500	33	145	" 36	2.4	60	-	146	165	"	112	330
"	"	"	"	F3-25	5,000	33	145	" 36	4.4	58	-	135	171	"	110	335
"	"	"	"	F3-26	5,000	34	145	" 36	1.5	185	-	129	139	"	92	335
"	"	"	"	F3-27	5,500	33	145	" 36	4.9	65	-	123	160	"	94	335
"	"	"	"	F3-28	6,000	33	145	" 36	3.8	70	-	105	128	"	87	335
"	"	"	"	F3-29	6,000	33	145	" 37	2.0	145	-	92	102	"	76	335
"	"	"	"	F3-30	6,000	33	145	" 36	3.6	60	-	94	113	"	77	335
"	"	"	"	F3-31	6,000	33	146	" 36	5.9	65	-	90	125	"	85	335
"	"	"	"	F3-32	5,500	33	146	" 36	5.9	50	-	87	121	"	76	340
"	"	"	"	F3-33	6,000	33	145	" 36	4.5	55	-	85	108	"	65	340
"	"	"	"	F3-34	5,800	33	145	" 36	4.2	58	-	86	108	"	65	340
"	"	"	"	F3-35	5,500	33	145	" 36	4.8	55	-	86	111	"	70	340
"	"	"	"	F3-36	6,000	33	145	" 36	3.2	60	-	91	107	"	66	340

Table I. Field Test Experimental Data (9/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^{0}F)	ATOMI-ZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^{0}F)
3-75	Firetube	No.5	CAS	F3-37	5,500	33	145	Air,36	4.8	55	-	89	115	NA	77	340
"	"	"	"	F3-38	5,500	34	145	" 37	4.2	56	-	88	110	"	72	340
"	"	"	"	F3-39	5,750	34	145	" 37	3.7	60	-	88	107	"	69	340
"	"	"	"	F3-40	6,000	34	145	" 37	5.3	55	-	87	116	"	81	340
"	"	"	"	F3-41	5,750	33	145	" 36	1.4	250	-	89	95	"	73	340
"	"	"	"	F3-42	6,000	34	150	" 36	3.3	75	-	108	128	"	84	340
"	"	"	"	F3-43	6,000	33	144	" 36	3.0	60	-	135	158	"	101	340
"	"	"	"	F3-44	7,000	32	145	" 35	7.0	70	-	134	201	"	147	340
"	"	"	"	F3-45	6,500	33	143	" 35	5.8	70	-	137	191	"	137	340
"	"	"	"	F3-46	6,500	33	144	" 35	5.0	70	-	140	184	"	129	340
"	"	"	"	F3-47	6,500	33	143	" 35	4.1	70	-	145	180	"	123	340
"	"	"	"	F3-48	5,500	33	142	" 35	1.9	75	-	146	161	"	109	340
"	"	"	"	F3-49	5,500	33	144	" 35	7.0	70	-	121	182	"	132	340
"	"	"	"	F3-50	5,500	33	143	" 35	6.0	70	-	125	175	"	123	340
"	"	"	"	F3-51	5,750	33	143	" 35	5.1	70	-	128	169	"	115	340
"	"	"	"	F3-52	6,000	32	145	" 35	4.0	70	-	130	161	"	108	340
"	"	"	"	F3-53	6,000	32	145	" 36	3.0	80	-	129	151	"	101	340
"	"	"	"	F3-54	6,500	33	143	" 35	2.1	180	-	123	137	"	96	340
"	"	"	"	F3-55	6,500	33	145	" 36	7.0	60	-	109	164	"	113	340
"	"	"	"	F3-56	6,000	33	144	" 26	6.0	60	-	106	148	"	103	340
"	"	"	"	F3-57	6,000	33	145	" 36	5.2	70	-	96	128	"	97	340
"	"	"	"	F3-58	6,250	33	145	" 36	4.0	90	-	88	109	"	88	340
"	"	"	"	F3-59	6,000	33	145	" 36	3.1	100	-	92	108	"	82	340
"	"	"	"	F3-60	6,000	33	145	" 38	2.1	168	-	88	98	"	76	340
"	"	"	"	F3-61	5,750	33	144	" 36	7.0	70	-	87	131	"	94	340
"	"	"	"	F3-62	6,500	33	144	" 36	6.1	70	-	81	114	"	86	340
"	"	"	"	F3-63	6,750	33	145	" 36	5.2	80	-	78	104	"	78	340

Table I. Field Test Experimental Data (10/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
3-75	Firetube	No.5	CAS	F3-64	6,000	33	145	Air,36	4.0	70	-	84	104	NA	69	340
"	"	"	"	F3-65	6,000	33	145	" 36	3.0	80	-	91	106	"	63	340
"	"	"	"	F3-66	6,000	33	146	" 36	6.2	60	-	142	202	"	142	335
"	"	"	"	F3-67	6,000	34	143	" 36	3.8	60	-	153	187	"	153	335
"	"	"	"	F3-68	5,500	34	145	" 36	2.2	65	-	157	175	"	111	330
"	"	"	"	F3-69	6,000	35	144	" 37	6.1	55	-	135	190	"	123	340
"	"	"	"	F3-70	5,000	35	144	" 37	4.0	60	-	148	183	"	107	340
"	"	"	"	F3-71	5,000	35	145	" 37	1.9	78	-	140	154	"	94	335
"	"	"	"	F3-72	5,500	35	146	" 37	6.1	50	-	125	176	"	103	340
"	"	"	"	F3-73	6,000	35	147	" 37	4.0	60	-	112	138	"	86	335
"	"	"	"	F3-74	5,500	35	146	" 37	2.0	243	-	105	116	"	74	335
"	"	"	"	F3-75	6,500	35	148	" 37	6.2	50	-	105	149	"	86	340
"	"	"	"	F3-76	5,000	35	148	" 37	4.0	60	-	98	121	"	69	345
"	"	"	"	F3-77	6,000	35	152	" 37	3.1	60	-	100	117	"	63	345
4-75	"	"	"	F3-78	5,500	34	146	" 36	5.8	25	-	136	188	"	136	-
"	"	"	"	F3-79	7,000	34	148	" 36	4.1	60	-	142	176	"	123	-
"	"	"	"	F3-80	7,500	34	145	" 36	2.1	70	-	145	161	"	110	-
"	"	"	"	F3-81	5,000	34	145	" 36	6.2	50	-	122	173	"	124	-
"	"	"	"	F3-82	6,000	34	145	" 36	3.8	65	-	117	143	"	106	-
"	"	"	"	F3-83	6,000	34	146	" 36	2.0	920	-	83	92	"	95	-
"	"	"	"	F3-84	6,000	34	148	" 36	6.0	90	-	82	115	"	102	-
"	"	"	"	F3-85	6,000	34	150	" 36	3.9	200	-	74	91	"	86	-
"	"	"	"	F3-86	6,000	34	150	" 36	2.0	180	-	82	91	"	74	-
"	"	"	"	F3-87	6,000	34	150	" 36	6.1	195	-	71	100	"	85	-
"	"	"	"	F3-88	6,500	34	149	" 36	4.0	80	-	80	99	"	68	-
"	"	"	"	F3-89	6,000	34	148	" 36	2.5	85	-	93	106	"	59	-
"	"	"	"	F3-90	6,000	34	148	" 36	4.1	160	-	76	94	"	87	-

Table I. Field Test Experimental Data (11/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMI-ZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
4-75	Firetube	No.5	CAS	F3-91	6,000	34	148	Air,36	4.1	335	-	76	94	NA	87	-
"	"	"	"	F3-92	6,000	34	148	" 36	6.0	70	-	77	108	19	102	-
"	"	"	"	F3-93	6,000	34	149	" 36	4.0	90	-	71	88	20	87	-
"	"	"	"	F3-94	6,000	34	148	" 36	6.0	60	-	78	109	23	102	-
"	"	"	"	F3-95	6,000	33	150	" 36	4.0	70	-	77	95	22	87	-
"	"	"	"	F3-96	6,500	34	150	" 36	6.2	70	-	75	106	34	105	-
"	"	"	"	F3-97	6,500	34	150	" 36	4.5	75	-	65	83	41	90	-
4-75	Firetube	Natural Gas	CAS	F3-98	2,500	-	-	NA	10.1	80	-	38	73	NA	182	
"	"	"	"	F3-99	1,500	-	-	"	7.0	107	-	44	66	"	145	
"	"	"	"	F3-100	1,500	-	-	"	4.2	267	-	48	60	"	123	
"	"	"	"	F3-101	1,500	-	-	"	2.6	1000+	-	47	54	"	113	
"	"	"	"	F3-102	4,000	-	-	"	7.6	802	-	32	50	"	129	
"	"	"	"	F3-103	8,500	-	-	"	5.4	80	-	63	85	"	131	
"	"	"	"	F3-104	8,500	-	-	"	3.0	83	-	70	82	"	115	
"	"	"	"	F3-105	8,500	-	-	"	4.6	87	-	64	82	"	113	
"	"	"	"	F3-106	8,500	-	-	"	3.1	255	-	70	82	"	103	
"	"	"	"	F3-107	7,750	-	-	"	5.4	167	-	56	78	"	101	
"	"	"	"	F3-108	7,750	-	-	"	2.9	1000+	-	64	74	"	84	
"	"	"	"	F3-109	8,000	-	-	"	5.4	1000+	-	59	79	"	90	
"	"	"	"	F3-110	9,000	-	-	"	2.8	590	-	68	78	"	71	
"	"	"	"	F3-111	9,000	-	-	"	5.2	120	-	62	82	"	71	
"	"	"	"	F3-112	8,500	-	-	"	2.9	185	-	69	79	"	57	
"	"	"	"	F3-113	6,750	-	-	"	5.5	70	-	63	85	"	132	
"	"	"	"	F3-114	7,750	-	-	"	2.7	315	-	71	81	"	113	
"	"	"	"	F3-115	7,750	-	-	"	5.3	105	-	61	82	"	118	

Table I. Field Test Experimental Data (12/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
4-75	Firetube	Natural Gas	CAS	F3-116	7,750	-	-	NA	2.6	1000+	-	69	79	NA	101	
"	"	"	"	F3-117	8,500	-	-	"	5.4	335	-	60	81	"	106	
"	"	"	"	F3-118	8,500	-	-	"	3.0	1000+	-	50	82	"	91	
"	"	"	"	F3-119	8,750	-	-	"	5.4	135	-	65	88	"	96	
"	"	"	"	F3-120	9,750	-	-	"	2.9	450	-	84	97	"	77	
"	"	"	"	F3-121	8,750	-	-	"	5.4	110	-	74	100	"	74	
"	"	"	"	F3-122	8,000	-	-	"	2.8	140	-	94	108	"	55	
"	"	"	"	F3-123	8,000	-	-	"	2.9	1000+	-	68	79	"	87	
"	"	"	"	F3-124	8,000	-	-	"	5.0	120	-	66	87	"	117	
"	"	"	"	F3-125	10,500	-	-	"	4.8	82	-	71	92	"	126	
"	"	"	"	F3-126	10,250	-	-	"	2.9	555	-	77	89	"	115	
"	"	"	"	F3-127	10,000	-	-	"	4.7	325	-	72	93	"	117	
"	"	"	"	F3-128	10,000	-	-	"	2.8	1000+	-	78	90	"	105	
"	"	"	"	F3-129	3,500	-	-	"	5.2	535	-	63	84	"	107	
"	"	"	"	F3-130	3,000	-	-	"	2.6	1000+	-	77	88	"	38	
"	"	"	"	F3-131	8,000	-	-	"	4.7	90	-	90	116	"	79	
"	"	"	"	F3-132	8,000	-	-	"	2.7	220	-	108	124	"	68	
"	"	"	"	F3-133	10,750	-	-	"	2.8	225	-	79	91	"	114	
"	"	"	"	F3-134	8,000	-	-	"	5.3	70	-	71	95	"	130	
"	"	"	"	F3-135	8,000	-	-	"	5.2	85	-	69	92	"	119	
"	"	"	"	F3-136	8,500	-	-	"	2.7	1000+	-	76	87	"	102	
"	"	"	"	F3-137	8,500	-	-	"	5.4	270	-	64	86	"	106	
"	"	"	"	F3-138	8,500	-	-	"	2.7	1000+	-	74	85	"	88	
"	"	"	"	F3-139	8,000	-	-	"	5.3	110	-	70	94	"	93	
"	"	"	"	F3-140	8,000	-	-	"	2.6	650	-	84	96	"	75	
"	"	"	"	F3-141	8,000	-	-	"	5.2	90	-	81	108	"	73	
"	"	"	"	F3-142	8,000	-	-	"	2.8	100	-	86	99	"	54	

Table I. Field Test Experimental Data (13/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMI-ZATION FLUID. PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
5-75	Firetube	Natural Gas	EAV,FGR	F4-1	4,750	-	-	NA	6.3	103	-	55	78	0	NA	350
"	"	"	"	F4-2	4,750	-	-	"	4.7	110	-	54	70	0	"	340
"	"	"	"	F4-3	4,750	-	-	"	2.8	123	-	58	67	0	"	340
"	"	"	"	F4-4	4,750	-	-	"	1.0	1000+	-	59	62	0	"	340
"	"	"	"	F4-5	4,750	-	-	"	5.1	110	-	57	75	0	"	340
"	"	"	"	F4-6	4,750	-	-	"	5.3	110	-	35	47	15	"	350
"	"	"	"	F4-7	4,750	-	-	"	4.2	105	-	35	44	16	"	345
"	"	"	"	F4-8	5,250	-	-	"	2.9	110	-	38	44	16	"	345
"	"	"	"	F4-9	5,250	-	-	"	0.9	1000+	-	34	35	18	"	345
"	"	"	"	F4-10	3,500	-	-	"	6.0	100	-	20	28	29	"	345
"	"	"	"	F4-11	3,500	-	-	"	4.3	100	-	21	27	33	"	345
"	"	"	"	F4-12	5,250	-	-	"	3.2	100	-	24	28	31	"	350
"	"	"	"	F4-13	5,750	-	-	"	1.2	500	-	20	21	31	"	350
"	"	"	"	F4-14	5,750	-	-	"	3.3	118	-	12	15	47	"	350
"	"	"	"	F4-15	6,000	-	-	"	0.7	1000+	-	7	7	52	"	355
"	"	"	"	F4-16	6,000	-	-	"	0.3	1000+	-	8	8	53	"	355
"	"	"	"	F4-17	5,250	-	-	"	4.5	108	-	13	17	49	"	355
"	"	"	"	F4-18	5,750	-	-	"	3.2	120	-	55	65	0	"	345
"	"	"	"	F4-19	8,750	-	-	"	1.9	155	-	76	83	0	"	355
"	"	"	"	F4-20	7,540	-	-	"	0.7	1000+	-	76	76	0	"	355
"	"	"	"	F4-21	7,250	-	-	"	3.7	30	-	77	93	0	"	355
"	"	"	"	F4-22	7,250	-	-	"	5.9	20	-	74	102	0	"	365
"	"	"	"	F4-23	8,500	-	-	"	5.1	20	-	56	74	8	"	370
"	"	"	"	F4-24	9,000	-	-	"	4.1	20	-	58	72	9	"	370
"	"	"	"	F4-25	9,000	-	-	"	2.1	75	-	57	63	11	"	360
"	"	"	"	F4-26	8,000	-	-	"	2.9	765	-	55	58	11	"	360
"	"	"	"	F4-27	8,000	-	-	"	5.0	30	-	31	40	25	"	365

Table I. Field Test Experimental Data (14/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
5-75	Firetube	Natural Gas	EAV,FGR	F4-28	7,250	-	-	NA	3.8	30	-	28	34	30	NA	350
"	"	"	"	F4-29	7,250	-	-	"	1.9	1000+	-	26	29	36	"	350
"	"	"	"	F4-30	7,250	-	-	"	1.0	200	-	24	25	35	"	350
"	"	"	"	F4-31	7,250	-	-	"	3.8	23	-	17	21	47	"	355
"	"	"	"	F4-32	7,250	-	-	"	3.1	20	-	18	21	47	"	-
"	"	"	"	F4-33	8,000	-	-	"	1.9	329	-	15	16	52	"	357
"	"	"	"	F4-34	8,000	-	-	"	1.0	1000+	-	12	12	55	"	356
"	"	"	"	F4-35	8,000	-	-	"	1.1	1000+	-	13	13	55	"	355
"	"	"	"	F4-36	6,500	-	-	"	2.3	-	-	72	80	0	"	-
10-75	Watertube	Natural Gas	EAV,FGR	W3-1	7,800	-	-	"	0.3	1000+	-	53	54	0	NA	380
"	"	"	"	W3-2	7,800	-	-	"	1.3	140	-	51	54	0	"	380
"	"	"	"	W3-3	8,000	-	-	"	2.5	10	-	54	61	0	"	385
"	"	"	"	W3-4	8,000	-	-	"	3.5	10	-	55	66	0	"	390
"	"	"	"	W3-5	7,000	-	-	"	2.3	13	-	28	31	13	"	380
"	"	"	"	W3-6	7,250	-	-	"	1.4	70	-	26	28	12	"	380
"	"	"	"	W3-7	7,000	-	-	"	3.6	25	-	30	36	11	"	380
"	"	"	"	W3-8	7,000	-	-	"	3.7	30	-	19	23	23	"	390
"	"	"	"	W3-9	7,000	-	-	"	2.6	30	-	16	18	25	"	390
"	"	"	"	W3-10	7,000	-	-	"	1.2	55	-	14	15	26	"	390
"	"	"	"	W3-11	9,750	-	-	"	3.1	20	-	50	54	0	"	400
"	"	"	"	W3-12	9,500	-	-	"	1.0	565	-	48	50	0	"	405
"	"	"	"	W3-13	9,500	-	-	"	2.6	20	-	18	21	22	"	410
"	"	"	"	W3-14	9,500	-	-	"	1.2	370	-	16	16	23	"	410

Table I. Field Test Experimental Data (15/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
3/76	Watertube	No. 5	FGR	W4-1	5,800	10	143	Steam, 34	7.3	10	-	110	169	0	NA	385
"	"	"	"	W4-2	7,000	12	144	" 33	5.2	10	-	105	40	20	"	400
"	"	"	"	W4-3	6,500	14	144	" 35	4.6	10	-	115	147	0	"	390
"	"	"	"	W4-4	8,000	14	145	" 34	4.3	10	-	115	145	14	"	390
"	"	"	"	W4-5	7,500	13	144	" 34	4.5	10	-	110	140	21	"	395
"	"	"	"	W4-6	7,500	13	145	" 35	4.3	10	-	105	132	31	"	400
"	"	"	EAV	W4-7	10,000	45	142	" 41	5.4	10	-	125	168	NA	"	415
"	"	"	"	W4-8	10,000	45	142	" 41	4.4	10	-	129	163	"	"	415
"	"	"	"	W4-9	10,000	45	142	" 41	3.5	20	-	125	150	"	"	415
"	"	"	"	W4-10	10,000	45	142	" 41	2.6	25	-	108	123	"	"	410
"	"	"	"	W4-11	10,000	45	142	" 41	1.8	825	-	95	104	"	"	400
"	"	"	FGR	W4-12	10,500	55	144	" 42	5.8	5	-	123	170	0	"	420
"	"	"	"	W4-13	10,000	55	144	" 42	3.3	20	-	119	141	10	"	410
"	"	"	"	W4-14	10,000	55	144	" 42	2.8	30	-	110	127	18	"	415
"	"	"	"	W4-15	9,500	55	142	" 40	3.3	35	-	112	133	20	"	415
"	"	"	"	W4-16	10,000	55	142	" 42	3.2	40	-	105	124	27	"	425
"	"	"	"	W4-17	10,500	55	142	" 42	3.1	38	-	118	138	0	"	415
3/76	Firetube	No. 5	EAV	F5-1	6,000	45	150	Air, 45	7.7	20	-	141	222	NA	"	370
"	"	"	"	F5-2	6,200	45	150	" 45	5.7	20	-	146	200	"	"	368
"	"	"	"	F5-3	6,500	45	150	" 45	3.3	20	-	149	177	"	"	362
"	"	"	"	F5-4	4,500	45	150	" 45	2.2	25	-	146	163	"	"	-
"	"	"	"	F5-5	5,000	45	150	" 45	3.9	30	-	146	179	"	"	364
"	"	"	"	F5-6	5,000	45	150	" 40	2.3	-	-	145	163	"	"	355
"	"	"	FGR	F5-7	5,000	45	150	" 40	3.8	40	-	147	180	0	"	-
"	"	"	"	F5-8	5,250	45	150	" 45	3.8	40	-	123	150	17	"	364
"	"	"	"	F5-9	5,700	45	150	" 45	3.9	40	-	119	146	30	"	374

Table I. Field Test Experimental Data (16/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^{o}F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^{o}F)
3/76	Firetube	No.5	FGR	F5-10	5,500	45	150	Air,45	3.9	40	-	113	137	36	NA	375
4/76	"	"	EAV	F5-11	9,000	60	150	" 55	6.0	40	-	167	233	NA	"	393
"	"	"	"	F5-12	8,000	60	150	" 55	5.2	40	-	169	224	"	"	388
"	"	"	"	F5-13	8,300	60	145	" 55	4.6	40	-	166	212	"	"	384
"	"	"	"	F5-14	8,500	60	145	" 55	3.6	40	-	164	198	"	"	381
"	"	"	"	F5-15	8,300	60	145	" 55	2.8	40	-	165	190	"	"	374
"	"	"	"	F5-16	8,500	60	143	" 55	1.6	60	-	159	172	"	"	372
"	"	"	"	F5-17	8,500	60	142	" 55	1.1	180	-	156	165	"	"	366
"	"	"	"	F5-18	8,500	60	142	" 55	1.4	70	-	158	169	"	"	368
"	"	"	FGR	F5-19	9,000	60	150	" 55	3.4	40	-	168	200	0	"	390
"	"	"	"	F5-20	9,000	60	150	" 55	3.5	40	-	150	180	8	"	390
"	"	"	"	F5-21	9,000	60	150	" 55	3.5	40	-	140	168	15	"	342
"	"	"	"	F5-22	8,000	60	148	" 55	3.2	60	-	141	166	19	"	394
"	"	"	EAV	F5-23	8,250	30	142	" 25	4.7	<10	-	170	219	NA	"	395
"	"	"	"	F5-24	8,250	29	140	" 25	5.6	20	-	174	237	"	"	400
"	"	"	"	F5-25	8,250	27	141	" 24	3.4	20	-	168	200	"	"	389
"	"	"	"	F5-26	9,250	30	142	" 25	2.3	30	-	172	193	"	"	393
"	"	"	"	F5-27	9,500	32	145	" 25	1.7	85	-	169	184	"	"	394
"	"	"	FGR	F5-28	9,000	30	142	" 25	3.1	30	-	155	182	8	"	400
"	"	"	"	F5-29	9,250	30	144	" 25	3.4	30	-	152	182	15	"	406
"	"	"	"	F5-30	6,000	22	144	" 22	3.7	30	-	167	202	0	"	378
"	"	"	"	F5-31	5,000	21	144	" 21	3.9	30	-	137	168	17	"	385
"	"	"	"	F5-32	6,250	23	142	" 23	4.0	30	-	131	162	30	"	379
"	"	"	"	F5-33	6,500	23	140	" 23	3.9	30	-	128	157	35	"	380

Table I. Field Test Experimental Data (17/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	φ (%)	STACK TEMP. (^oF)
4/76	Firetube	Methanol	EAV	F6-1	5,800	22	50	Air,23	8.9	30	-	30	52	NA	NA	362
"	"	"	"	F6-2	2,700	22	50	" 23	4.0	30	-	42	52	"	"	355
"	"	"	"	F6-3	2,700	22	50	" 23	2.7	30	-	45	52	"	"	-
"	"	"	"	F6-4	4,300	21	51	" 23	2.7	50	-	44	51	"	"	350
"	"	"	"	F6-5	4,000	21	52	" 23	0.7	55	-	44	46	"	"	350
"	"	"	"	F6-6	4,700	21	52	" 23	0.3	150	-	43	44	"	"	350
"	"	"	"	F6-7	6,500	28	52	" 27	8.8	43	-	28	48	"	"	376
"	"	"	"	F6-8	7,000	27	52	" 27	4.9	40	-	45	59	"	"	367
"	"	"	"	F6-9	6,000	28	52	" 27	3.2	50	-	46	54	"	"	366
"	"	"	"	F6-10	5,200	28	52	" 27	1.7	60	-	46	50	"	"	363
"	"	"	"	F6-11	6,000	28	53	" 27	0.7	400	-	45	47	"	"	363
"	"	"	"	F6-12	7,100	28	53	" 27	0.3	1000+	-	45	46	"	"	360
"		"	FGR	F6-13	6,700	27	53	" 27	4.0	45	-	46	57	0	"	365
"	"	"	"	F6-14	6,200	28	54	" 27	4.0	65	-	18	22	17	"	372
"	"	"	"	F6-15	6,100	28	54	" 27	4.0	50	-	9	11	33	"	376
"	"	"	"	F6-16	5,900	30	54	" 28	4.0	53	-	7	9	39	"	382
"	"	"	"	F6-17	6,500	28	54	" 27	4.0	55	-	5	6	52	"	382
"	"	"	EAV	F6-18	11,000	50	54	" 40	6.3	55	-	41	59	"	"	398
"	"	"	"	F6-19	10,000	50	53	" 40	4.2	50	-	47	59	"	"	395
"	"	"	"	F6-20	9,800	50	53	" 40	2.2	70	-	50	56	"	"	395
"	"	"	"	F6-21	9,200	50	53	" 40	1.2	480	-	50	53	"	"	395
"	"	"	"	F6-22	7,600	62	60	" 54	3.6	48	-	36	43	"	"	350
"	"	"	"	F6-23	4,100	62	60	" 54	9.0	43	-	25	44	"	"	358
"	"	"	"	F6-24	3,600	62	60	" 54	4.5	40	-	35	45	"	"	350
"	"	"	"	F6-25	4,000	62	60	" 54	1.6	43	-	37	40	"	"	343
"	"	"	"	F6-26	4,600	62	60	" 54	0.3	900	-	37	38	"	"	341
"	"	"	"	F6-27	6,000	60	74	" 60	5.7	45	-	38	52	"	"	360

Table I. Field Test Experimental Data (18/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
4/76	Firetube	Methanol	EAV	F6-28	5,700	72	60	Air,60	8.4	40	-	32	53	NA	NA	358
"	"	"	"	F6-29	5,700	74	60	" 60	3.9	60	-	44	54	"	"	351
"	"	"	"	F6-30	5,900	74	60	" 60	2.4	68	-	45	51	"	"	346
"	"	"	"	F6-31	5,900	74	60	" 60	0.6	1000+	-	49	51	"	"	344
"	"	"	"	F6-32	6,000	75	65	" 60	9.4	45	-	25	45	"	"	365
"	"	"	"	F6-33	6,000	75	65	" 60	6.0	45	-	35	49	"	"	356
"	"	"	"	F6-34	5,500	75	65	" 60	4.2	40	-	37	46	"	"	350
"	"	"	"	F6-35	5,500	75	65	" 60	2.7	35	-	39	45	"	"	348
"	"	"	"	F6-36	5,500	75	65	" 60	1.2	360	-	39	41	"	"	342
"	"	"	"	F6-37	5,500	75	66	" 60	0.3	1000+	-	39	40	"	"	340
"	"	"	FGR	F6-38	6,000	75	66	" 60	3.9	40	-	15	19	19	"	360
"	"	"	"	F6-39	5,000	75	66	" 60	4.0	40	-	8	10	32	"	360
"	"	"	"	F6-40	6,000	75	66	" 60	4.0	40	-	6	7	36	"	360
"	"	"	"	F6-41	6,000	75	66	" 60	3.9	40	-	5	6	40	"	360
"	"	"	EAV	F6-42	4,000	84	70	" 54	8.6	33	-	25	42	NA	"	358
"	"	"	"	F6-43	4,000	62	68	" 54	4.4	40	-	36	46	"	"	351
"	"	"	"	F6-44	4,000	62	68	" 54	1.4	38	-	37	40	"	"	346
"	"	"	FGR	F6-45	9,300	62	68	" 40	4.2	50	-	47	59	0	"	395
"	"	"	"	F6-46	9,300	62	68	" 40	4.1	45	-	26	32	8	"	395
"	"	"	"	F6-47	9,300	62	68	" 40	4.0	48	-	19	23	14	"	400
"	"	"	"	F6-48	9,300	64	68	" 40	4.1	50	-	17	21	17	"	400
"	"	"	"	F6-49	9,300	62	68	" 40	2.3	220	-	8	9	27	"	400
"	"	"	"	F6-50	9,300	62	68	" 40	2.2	58	-	12	13	21	"	400
"	"	"	"	F6-51	9,300	62	68	" 40	2.2	53	-	15	17	17	"	400
"	"	"	"	F6-52	9,300	62	68	" 40	2.2	72	-	23	26	11	"	396
"	"	"	FT	F6-53	9,300	62	68	" 40	1.8	180	-	48	53	0	"	392
"	"	"	"	F6-54	5,600	50	106	" 34	8.9	53	-	17	30	NA	"	378

Table I. Field Test Experimental Data (19/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
4/76	Firetube	Methanol	FT	F6-55	6,000	50	102	Air, 34	6.8	35	-	36	53	NA	NA	370
"	"	"	"	F6-56	6,000	50	100	" 34	3.0	40	-	43	50	"	"	364
"	"	"	"	F6-57	6,200	50	102	" 34	1.4	65	-	44	47	"	"	358
"	"	"	"	F6-58	6,300	50	102	" 34	0.8	220	-	44	46	"	"	358
"	"	"	"	F6-59	6,300	50	102	" 34	5.1	40	-	42	55	"	"	360
"	"	"	"	F6-60	6,100	50	74	" 30	3.6	45	-	48	58	"	"	365
"	"	"	"	F6-61	6,800	50	72	" 30	8.3	45	-	28	46	"	"	378
"	"	"	"	F6-62	6,800	50	72	" 32	0.7	1000+	-	43	44	"	"	362
"	"	"	"	F6-63	6,800	50	72	" 32	1.7	50	-	46	50	"	"	362
"	"	"	"	F6-64	6,000	75	70	" 60	8.7	50	-	23	39	"	"	341
"	"	"	"	F6-65	6,000	75	72	" 60	4.0	50	-	34	42	"	"	335
"	"	"	"	F6-66	5,000	75	70	" 60	1.6	65	-	35	38	"	"	342
"	"	"	"	F6-67	5,000	75	70	" 60	0.7	630	-	35	36	"	"	332
"	"	"	"	F6-68	5,200	75	108	" 60	5.4	45	-	31	42	"	"	335
"	"	"	"	F6-69	5,200	75	108	" 60	8.3	45	-	24	40	"	"	341
"	"	"	"	F6-70	5,000	75	108	" 60	3.7	45	-	34	41	"	"	341
"	"	"	"	F6-71	5,000	75	106	" 60	0.8	560	-	35	36	"	"	321
"	"	"	"	F6-72	5,000	75	108	" 60	1.5	73	-	36	39	"	"	321
"	"	"	"	F6-73	4,400	39	100	" 25	8.2	30	-	28	46	"	"	352
"	"	"	"	F6-74	4,400	39	100	" 25	4.4	50	-	36	46	"	"	346
"	"	"	"	F6-75	4,500	39	98	" 25	3.2	50	-	37	44	"	"	342
"	"	"	"	F6-76	4,100	39	98	" 25	0.6	100	-	37	38	"	"	336
"	"	"	"	F6-77	4,000	39	96	" 25	1.8	65	-	38	42	"	"	336
"	"	"	"	F6-78	4,000	38	68	" 24	1.4	45	-	40	43	"	"	341
"	"	"	"	F6-79	4,000	38	68	" 23	0.8	45	-	39	81	"	"	340
"	"	"	"	F6-80	3,500	38	66	" 23	3.4	30	-	41	49	"	"	330
"	"	"	"	F6-81	3,500	38	66	" 23	5.1	30	-	39	52	"	"	330
"	"	"	"	F6-82	3,500	38	66	" 23	8.3	35	-	32	53	"	"	345

Table I. Field Test Experimental Data (20/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
4/76	Watertube	Methanol	EAV	W5-1	9,000	27	82	Air, 32	.8	1000+	-	15	16	NA	NA	380
"	"	"	"	W5-2	9,000	27	82	" 33	1.6	625	-	15	16	"	"	380
"	"	"	"	W5-3	9,000	27	84	" 34	3.2	100	-	16	19	"	"	380
"	"	"	"	W5-4	8,500	27	84	" 35	5.3	30	-	16	21	"	"	385
"	"	"	"	W5-5	8,600	28	88	" 37	8.4	60	-	14	23	"	"	400
"	"	"	RO,100%	W5-6	8,700	26	84	" 32	0.9	1000+	-	15	16	"	"	375
"	"	"	" 94	W5-7	9,000	27	84	" 32	1.2	1000+	-	15	16	"	"	375
"	"	"	" 80	W5-8	9,000	26	84	" 32	1.1	1000+	-	16	17	"	"	375
"	"	"	" 81	W5-9	9,200	26	84	" 34	1.3	450	-	17	18	"	"	375
"	"	"	" 75	W5-10	9,200	26	84	" 34	1.3	130	-	17	18	"	"	375
"	"	"	" 69	W5-11	9,000	26	84	" 34	1.3	105	-	17	18	"	"	375
"	"	"		W5-12	9,000	26	84	" 36	1.1	280	-	16	17	"	"	375
"	"	"		W5-13	8,400	26	84	" 36	1.3	220	-	14	15	"	"	375
"	"	"		W5-14	8,500	26	84	" 35	1.1	555	-	13	14	"	"	375
"	"	"		W5-15	9,500	30	85	" 38	1.3	1000+	-	11	12	"	"	380
"	"	"		W5-16	9,500	30	84	" 40	1.4	1000+	-	11	12	"	"	380
"	"	"	FGR	W5-17	9,000	26	85	" 34	3.4	70	-	16	19	0	"	380
"	"	"	"	W5-18	9,200	28	85	" 35	3.3	40	-	9	11	11	"	390
"	"	"	"	W5-19	9,000	28	85	" 35	2.2	55	-	9	10	12	"	390
"	"	"	"	W5-20	9,000	28	85	" 35	2.1	60	-	7	8	17	"	390
"	"	"	"	W5-21	9,000	28	85	" 35	3.3	50	-	7	8	16	"	395
"	"	"	"	W5-22	9,000	28	85	" 36	3.4	50	-	6	7	23	"	400
"	"	"	"	W5-23	9,000	28	85	" 36	2.2	50	-	6	7	25	"	400
"	"	"	"	W5-24	9,000	28	85	" 36	2.0	55	-	5	6	32	"	400
"	"	"	"	W5-25	9,000	28	86	" 36	3.3	50	-	5	6	30	"	400
"	"	"	"	W5-26	8,600	28	86	" 37	3.6	65	-	5	6	34	"	405
"	"	"	AP	W5-27	8,700	27	75	Steam,26	5.3	40	-	16	21	NA	"	380

Table I. Field Test Experimental Data (21/23)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID. PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
4/76	Watertube	Methanol	AP	W5-28	9,000	27	75	Steam,26	0.7	1000+	-	15	16	NA	NA	375
"	"	"	"	W5-29	8,700	30	75	" 33	1.0	670	-	15	16	"	"	375
"	"	"	"	W5-30	8,700	30	75	" 33	5.0	55	-	17	22	"	"	380
"	"	"	"	W5-31	8,800	32	75	" 40	5.5	60	-	17	23	"	"	380
"	"	"	"	W5-32	8,500	37	76	" 40	5.4	50	-	18	24	"	"	390
"	"	"	"	W5-33	8,700	38	76	" 48	5.5	50	-	19	26	"	"	390
"	"	"	"	W5-34	8,900	40	76	" 56	5.2	50	-	17	23	"	"	395
"	"	"	"	W5-35												
"	"	"	"	W5-36	9,000	40	78	" 56	1.1	135	-	18	19	"	"	385
"	"	"	"	W5-37	9,200	37	78	" 48	0.8	90	-	20	21	"	"	385
"	"	"	"	W5-38	9,200	33	78	" 40	1.2	250	-	17	18	"	"	385
"	"	"	"	W5-39	10,000	30	78	" 33	1.0	565	-	14	15	"	"	380
"	"	"	"	W5-40	9,000	28	76	" 26	0.9	1000+	-	14	15	"	"	380
"	"	"	"	W5-41	9,000	25	76	" 20	1.1	1000+	-	15	16	"	"	375
"	"	"	"	W5-42	9,000	22	78	" 14	0.9	1000+	-	17	18	"	"	375
"	"	"	"	W5-43	9,000	22	78	" 14	5.3	160	-	13	7	"	"	390
"	"	"	"	W5-44	9,000	25	78	" 20	5.5	60	-	16	22	"	"	390
"	"	"	"	W5-45	8,000	40	78	Air, 30	5.0	180	-	9	12	"	"	385
"	"	"	"	W5-46	9,000	45	78	" 38	4.9	155	-	10	13	"	"	385
"	"	"	"	W5-47	9,000	50	78	" 46	5.1	45	-	13	17	"	"	385
"	"	"	"	W5-48	9,000	52	78	" 54	5.4	50	-	16	22	"	"	380
"	"	"	"	W5-49	9,000	52	78	" 57	5.2	45	-	17	23	"	"	380
"	"	"	"	W5-50	8,400	51	78	" 57	1.0	1000+	-	15	16	"	"	370
"	"	"	"	W5-51	8,300	50	78	" 52	0.9	1000+	-	14	15	"	"	370
"	"	"	"	W5-52	8,600	48	78	" 46	1.0	1000+	-	12	13	"	"	370
"	"	"	"	W5-53	8,700	44	78	" 38	1.0	1000+	-	8	8	"	"	375
"	"	"	"	W5-54	9,000	40	78	" 30	1.2	1000+	-	8	8	"	"	375

Table I. Field Test Experimental Data (22/23)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (^oF)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (^oF)
4/76	Watertube	Methanol	AP	W5-55	9,000	35	78	Air,24	1.1	1000+	-	8	8	NA	NA	375
"	"	"	"	W5-56	9,000	28	78	" 16	1.0	1000+	-	10	11	"	"	375
"	"	"	"	W5-57	8,400	28	78	" 16	5.1	590	-	11	15	"	"	380
"	"	"	"	W5-58	9,000	25	78	" 11	5.2	650	-	9	12	"	"	385
"	"	"	"	W5-59	8,700	35	78	" 24	5.2	595	-	9	12	"	"	385
"	"	"	"	W5-60	8,800	30	78	" 11	1.0	1000+	-	10	11	"	"	375
"	"	"	EAV	W5-61	11,500	40	62	" 18	6.1	350	-	11	16	"	"	442
"	"	"	"	W5-62	12,200	40	62	" 18	3.4	1000+	-	12	14	"	"	440
"	"	"	"	W5-63	11,400	40	62	" 18	2.0	1000+	-	11	12	"	"	438
"	"	"	"	W5-64	12,000	40	62	" 18	0.5	1000+	-	13	13	"	"	420
"	"	"	FGR	W5-65	10,700	40	62	" 18	1.9	1000+	-	7	8	9	"	440
"	"	"	"	W5-66	10,700	40	62	" 18	3.3	1000+	-	8	9	9	"	448
"	"	"	"	W5-67	11,400	40	64	" 18	3.4	1000+	-	5	6	12	"	452
"	"	"	"	W5-68	11,400	40	64	" 18	2.1	1000+	-	5	6	15	'	448
"	"	"	"	W5-69	12,000	40	62	" 18	2.2	1000+	-	4	4	20	"	448
"	"	"	"	W5-70	12,200	40	62	" 18	3.6	845	-	5	6	19	"	450
"	"	"	"	W5-71	11,500	40	62	" 18	3.6	720	-	4	5	24	"	455
"	"	"	"	W5-72	11,500	40	62	" 18	1.8	1000+	-	3	3	26	"	450
"	"	"	"	W5-73	11,500	40	62	" 18	2.0	1000+	-	3	3	19	"	450
"	"	"	AP	W5-74	11,700	38	62	" 12	5.4	750	-	7	9	NA	"	440
"	"	"	"	W5-75	11,800	40	62	" 16	5.2	450	-	11	15	"	"	440
"	"	"	"	W5-76	11,700	44	62	" 20	5.3	500	-	11	15	"	"	440
"	"	"	"	W5-77	11,800	46	64	" 24	5.3	675	-	10	13	"	"	440
"	"	"	"	W5-78	11,800	50	64	" 30	5.4	480	-	9	12	"	"	440
"	"	"	"	W5-79	11,500	42	65	Steam,16	5.3	755	-	8	11	"	"	450
"	"	"	"	W5-80	12,200	42	65	" 20	5.2	225	-	13	17	"	"	450
"	"	"	"	W5-81	12,000	45	65	" 24	5.1	95	-	15	20	"	"	450

Table I. Field Test Experimental Data (23/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
4/76	Watertube	Methanol	AP	W5-82	13,000	45	65	Steam,28	5.1	65	-	15	20	NA	NA	450
"	"	"	"	W5-83	11,800	48	65	" 33	5.1	60	-	15	20	"	"	450
"	"	"	"	W5-84	11,700	50	65	" 38	3.6	45	-	14	17	"	"	450
"	"	"	"	W5-85	12,000	52	65	" 47	5.2	40	-	15	20	"	"	450
"	"	"	EAV	W5-86	12,000	52	65	" 37	4.0	55	-	15	17	"	"	448
"	"	"	"	W5-87	11,700	50	62	" 37	3.0	120	-	14	16	"	"	445
"	"	"	"	W5-88	11,700	50	62	" 37	2.0	430	-	14	15	"	"	445
"	"	"	"	W5-89	12,200	50	62	" 37	1.8	960	-	14	15	"	"	443
"	"	"	FGR	W5-90	12,000	50	65	" 37	1.8	290	-	8	9	10	"	450
"	"	"	"	W5-91	12,000	50	65	" 37	3.3	55	-	8	9	9	"	452
"	"	"	"	W5-92	12,000	50	65	" 37	3.3	60	-	7	8	13	"	458
"	"	"	"	W5-93	12,000	50	65	" 37	1.9	230	-	7	8	14	"	458
"	"	"	"	W5-94	12,000	52	65	" 37	2.0	120	-	5	6	21	"	458
"	"	"	"	W5-95	12,000	52	65	" 37	3.3	40	-	6	7	20	"	458
"	"	"	"	W5-96	11,400	52	65	" 37	3.5	45	-	5	6	26	"	468
"	"	"	"	W5-97	11,500	52	65	" 37	1.8	135	-	4	4	27	"	468

Table I. Field Test Experimental Data (24/24)

TEST DATE	SYSTEM	FUEL	TEST TYPE	TEST NO.	LOAD (lbm/hr)	FUEL PRESS. (psig)	FUEL TEMP. (°F)	ATOMIZATION FLUID, PRESS. (psig)	O_2 (%)	CO (ppm)	CO_2 (%)	NO_x ppm, as meas.	NO_x, @ 0% O_2	FGR (mass %)	ϕ (%)	STACK TEMP. (°F)
4/76	Watertube	Natural Gas	EAV	W6-1	8,000	-	-	NA	4.9	10	-	55	72	NA	NA	428
"	"	"	"	W6-2	8,400	-	-	"	8.3	15	-	44	72	"	"	438
"	"	"	"	W6-3	8,400	-	-	"	3.2	20	-	56	65	"	"	430
"	"	"	"	W6-4	8,700	-	-	"	1.5	130	-	53	57	"	"	425
"	"	"	"	W6-5	9,300	-	-	"	1.1	420	-	53	56	"	"	426
"	"	"	RO,94	W6-6	9,200	-	-	"	1.3	150	-	54	58	"	"	422
"	"	"	" 86	W6-7	9,200	-	-	"	1.1	180	-	54	58	"	"	422
"	"	"	" 81	W6-8	9,200	-	-	"	1.0	100	-	54	58	"	"	420
"	"	"	" 75	W6-9	8,700	-	-	"	1.0	105	-	56	57	"	"	420
"	"	"	" 69	W6-10	8,700	-	-	"	.9	110	-	59	62	"	"	420
"	"	"	" 67	W6-11	9,000	-	-	"	1.1	25	-	62	65	"	"	422
"	"	"	" 56	W6-12	9,000	-	-	"	.8	60	-	64	67	"	"	419
"	"	"	" 51	W6-13	9,200	-	-	"	1.0	35	-	67	71	"	"	380
"	"	"		W6-14	9,000	-	-	"	1.3	20	-	75	79	"	"	375
"	Firetube	"	EAV	F7-1	7,200	-	-	"	8.1	40	-	68	110	"	"	372
"	"	"	"	F7-2	7,200	-	-	"	5.1	40	-	75	99	"	"	367
"	"	"	"	F7-3	7,200	-	-	"	4.1	40	-	76	94	"	"	368
"	"	"	"	F7-4	7,100	-	-	"	2.4	80	-	78	88	"	"	368
"	"	"	"	F7-5	6,500	-	-	"	1.6	230	-	77	84	"	"	368
"	"	"	FGR	F7-6	6,000	-	-	"	0.7	1000+	-	73	76	0	"	362
"	"	"	"	F7-7	6,000	-	-	"	4.0	40	-	44	54	12	"	374
"	"	"	"	F7-8	6,000	-	-	"	4.1	40	-	32	40	19	"	377
"	"	"	"	F7-9	6,000	-	-	"	4.1	40	-	26	32	26	"	378
"	"	"	"	F7-10	6,400	-	-	"	4.1	40	-	18	22	32	"	385
"	"	"	"	F7-11	6,400	-	-	"	4.1	40	-	17	21	38	"	385
"	"	"	"	F7-12	6,400	-	-	"	4.1	40	-	15	19	45	"	384

APPENDIX B. OIL-FIRED DISTRIBUTED MIXING BURNER TEST DATA

Table B-1.

Fuel: 0.77% Nitrogen
Primary Swirl: 45^{o}
Air Temperature: Ambient
Nozzle: Monarch
Secondary Swirl: 60^{o}
Primary Flow: 300 scfm
Divergent ${}^{L}/_{D}$ = 1
Staged Air Ducts: 16

Load $\times 10^6$ Btu/hr	Atomizing Fluid Pressure psig	Burner Zone Stoichiometry	Exhaust Composition (dry as measured) NO_x ppm	CO ppm	O_2 %
7.3	40	Unstaged	240	x	10.2
7.3	40	Unstaged	330	x	8.5
7.3	40	Unstaged	360	13	7.9
7.3	40	Unstaged	460	6	5.1
7.3	40	Unstaged	465	5	1.6
7.3	40	Unstaged	470	7	2.55
7.3	40	Unstaged	465	x	3.7
9.1	68	Unstaged	425	43	2.6
9.1	68	Unstaged	430	74	2.5
9.1	68	Unstaged	475	46	4.0
9.1	68	Unstaged	445	19	5.7
9.1	68	Unstaged	445	11	6.1
12.4	80	Unstaged	490	41	0.4
12.4	80	Unstaged	535	31	2.3
12.4	80	Unstaged	540	23	2.9
12.4	80	Unstaged	540	18	3.0

Table B-2.

Fuel: 0.77% Nitrogen

Nozzle: Monarch

Divergent $^L/_D$ = 1

Primary Swirl: 45°

Secondary Swirl: 60°

Staged Air Ducts: 16

Air Temperature: Ambient

Burner Zone Stoichiometry: Unstaged

Load x10^6 Btu/hr	Atomizing Fluid Pressure psig	Primary Flow scfm	Exhaust Composition (dry as measured)		
			NO_x ppm	CO ppm	O_2 %
12.4	80	445	555	80	3.7
12.4	80	300	600	78	3.0
12.4	80	200	580	83	3.0
12.4	80	100	590	80	3.1
7.2	40	100	450	79	2.6
7.2	40	200	445	62	4.9
7.2	40	300	480	73	2.7
7.2	40	400	565	77	2.7

Table B-3.

Fuel: 0.77% Nitrogen
Primary Swirl: 45^0
Air Temperature: Ambient
Nozzle: Monarch
Secondary Swirl: 60^0
Primary Flow: 400 scfm
Divergent $^L/_D = 1$
Staged Air Ducts:

Load $\times 10^6$ Btu/hr	Atomizing Fluid Pressure psig	Burner Zone Stoichiometry	Exhaust Composition (dry as measured) NO_x ppm	CO ppm	O_2 %
9.3	75	Unstaged	565	62	4.2
9.3	75	Unstaged	520	71	2.7
9.3	75	Unstaged	440	100	0.6
9.3	75	Unstaged	460	74	1.0
9.3	75	Unstaged	490	63	2.2

Table B-4.

Fuel: 0.71% Nitrogen　Primary Swirl: 45°　Air Temperature: Ambient

Nozzle: Monarch　Secondary Swirl: 60°　Primary Flow: 100 scfm

Divergent $^L/_D = 1$

Load x10^6 Btu/hr	Atomizing Fluid Pressure psig	Burner Zone Stoichiometry	Exhaust Composition (dry as measured)		
			NO_x ppm	CO ppm	O_2 %
9	70	1.15	535	35	4.9
9	70	1.15	485	36	6.5
9	70	1.15	440	30	8.3
9	70	1.15	420	30	9.0
9	70	1.0	235	160	12.5
9	70	1.0	275	55	10.25
9	70	1.0	295	43	8.4
9	70	1.0	320	42	7.05
9	70	1.0	325	49	4.75
9	70	1.0	330	60	3.4
9	70	1.0	280	164	2.3
9	70	0.85	215	95	10.5
9	70	0.85	230	71	8.1
9	70	0.85	240	66	5.9
9	70	0.85	225	95	4.2
9	70	0.85	225	93	3.9
9	70	0.70	185	92	12.0
9	70	0.70	170	150	11.0
9	70	0.70	165	160	10.9
9	70	0.70	160	180	9.5
9	70	0.60	120	133	11.2
9	70	0.60	120	285	10.3

Table B-5.

Fuel: 0.77% Nitrogen

Primary Swirl: 45^0

Nozzle: Monarch

Secondary Swirl: 60^0

Staged Air Ducts: 16

Load $\times 10^6$ Btu/hr	Atomizing Fluid Pressure psig	Burner Zone Stoichiometry	Exhaust Composition (dry as measured)		
			NO_x ppm	CO ppm	O_2 %
10	80	Unstaged	100	450	2.0
10	80	Unstaged	104	140	4.0
10	80	Unstaged	108	120	6.0
10	80	0.9	98	440	2.5
10	80	0.9	99	180	3.8
10	80	0.9	115	120	6.5
10	80	0.7	110	200	1.0
10	80	0.7	112	105	1.5
10	80	0.7	120	95	3.6
10	80	0.7	130	74	5.7
10	80	0.7	131	90	7.4
10	80	0.5	110	98	1.9
10	80	0.5	111	90	4.0
10	80	0.5	112	85	6.2

Table B-6.

Fuel: 0.77% Nitrogen

Nozzle: Monarch

Primary Swirl: 45°

Secondary Swirl: 60°

Staged Air Ducts: 8

Load $\times 10^6$ Btu/hr	Atomizing Fluid Pressure psig	Burner Zone Stoichiometry	Exhaust Composition (dry as measured)		
			NO_x ppm	CO ppm	O_2 %
9.4	70	1.13	342	13	6.2
9.4	70	1.12	340	13	5.2
9.4	70	1.0	317	x	5.9
9.4	70	1.0	315	93	4.9
9.4	70	0.95	240	120	6.6
9.4	70	0.93	243	202	4.9
9.4	70	0.74	241	230	3.6
9.4	70	0.75	250	74	4.8
9.4	70	0.68	230	80	5.3
9.4	70	0.60	197	156	5.0
9.4	70	0.51	159	265	5.2

Table B-7.

Fuel: 0.77% Nitrogen
Primary Swirl: 45°
Air Temperature: Ambient

Nozzle: Monarch
Secondary Swirl: 60°
Primary Flow: 400 scfm

Divergent $^L/_D$ = 1
Staged Air Ducts: 16

Load x10^6 Btu/hr	Atomizing Fluid Pressure psig	Burner Zone Stoichiometry	Exhaust Composition (dry as measured)		
			NO_x ppm	CO ppm	O_2 %
8.75	70	1.15	535	24	3.3
8.75	70	1.15	315	x	11.0
8.75	70	1.15	405	x	8.7
8.75	70	1.15	440	3	6.9
8.75	70	1.15	490	8	4.8
8.75	70	1.15	500	11	3.45
8.75	70	1.0	200	246	12.8
8.75	70	1.0	220	245	12.0
8.75	70	1.0	290	30	9.9
8.75	70	1.0	340	54	7.6
8.75	70	1.0	390	48	5.3
8.75	70	1.0	325	156	1.9
8.75	70	1.0	385	97	4.8
8.75	70	1.0	360	138	3.3
8.75	70	0.85	205	160	13.0
8.75	70	0.85	245	52	10.0
8.75	70	0.85	310	32	7.2
8.75	70	0.85	350	39	3.85
8.75	70	0.85	340	28	5.7
8.75	70	0.85	360	31	4.4
8.75	70	0.85	365	33	3.45
8.75	70	0.85	325	50	2.25

Table B-7, Con't.

Load $x10^6$ Btu/hr	Atomizing Fluid Pressure psig	Burner Zone Stoichiometry	Exhaust Composition (dry as measured)		
			NO_x ppm	CO ppm	O_2 %
8.75	70	0.70	200	45	11.7
8.75	70	0.70	230	49	8.3
8.75	70	0.70	225	108	6.8
8.75	70	0.70	215	210	5.0
8.75	70	0.70	210	300	3.2
8.75	70	0.60	225	65	12.2
8.75	70	0.60	230	37	10.0
8.75	70	0.60	215	52	7.2

Table B-8.

Fuel: 0.77% Nitrogen
Primary Swirl: 45°
Nozzle: Monarch
Secondary Swirl: 60°
Primary Flow: 400 scfm
Divergent $^L/_D$ = 1
Staged Air Ducts: 16

Load x10^6 Btu/hr	Atomizing Fluid Press. psig	Burner Zone Stoichiometry	Air Preheat °C	Exhaust Composition (dry as measured)		
				NO_x ppm	CO ppm	O_2 %
6.7	40	1.23	25	515	83	6.1
6.7	40	1.16	26	560	71	3.1
6.7	40	1.16	26	500	67	5.3
6.7	40	0.99	27	380	47	5.1
11.0	80	1.21	24	480	84	3.75
11.0	80	1.03	23	410	78	4.65
11.0	80	1.03	24	395	73	6.05
11.0	80	0.87	25	290	75	6.2
11.0	80	0.87	87	275	83	3.4
11.0	80	0.73	28	160	550	4.5
9.4	70	1.07	189	450	11	1.3
9.4	70	1.07	193	425	10	3.5
9.4	70	1.07	195	405	9	5.05
9.4	70	0.86	185	315	10	3.3
9.4	70	0.86	178	310	9	4.2
9.4	70	0.73	169	230	14	5.6
9.4	70	0.73	171	250	12	4.7
6.8	40	1.22	157	530	100	3.9
6.8	40	122	159	485	100	4.3
6.8	40	1.22	158	445	100	5.6
6.8	40	1.05	158	435	90	5.1
6.8	40	0.89	159	320	90	4.2
6.8	40	0.89	158	300	93	5.25
6.8	40	0.78	155	275	119	4.7
6.8	40	0.78	157	195	134	5.5